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DYNAMIC BALANCING.

BY COMMANDER F. J. CLEARY, U. S. N., MEMBER.

A detailed description of the method developed and used at Mare Island for dynamically balancing the rotating elements (four rotors, one large reduction gear and two propellers) of the turbine propelling machinery U. S. S. "Shaw."

I.—PRELIMINARY.

When the building of the *Shaw* was awarded to Mare Island, it was desired to make a record in excellence of construction, both in hull and machinery. As Shop Superintendent of the Machinery Division I had direct charge of the building of the turbines, a type of propelling machinery with which Mare Island had no experience. From a careful study of various articles and books, and discussion with engineers having practical experience underway with marine turbine installations, I obtained the information that in a very great many marine turbine installations there were certain critical speeds at which the vibration of the turbines was excessive. This, regardless of the fact that all rotating elements had been

very carefully balanced statically. To cite a particular case, that of a rotor designed for a maximum speed of 670 r.p.m., there was no apparent vibration at any speed of rotation less than 225 r.p.m. As soon as the speed exceeded 225 revolutions, vibration began, rapidly increasing to a maximum at about 500 revolutions (at which speed the vibration was violent) and gradually decreasing to 670 revolutions, at which speed there still remained considerable vibration. In other words, this carefully statically balanced rotor was dynamically unbalanced for all speeds of rotation from 225 to 670 revolutions per minute. While it is a fact that the rotor bearings would prevent *free* vibration of the *rotor,* the dynamic unbalance would exist and would result in a thrust delivered all around the circumference of the bearings for every revolution. In destroyers, where every pound of weight possible is eliminated, I was convinced that such a condition could very readily cause vibration throughout the ship, affecting the gun platform, causing undue wear of bearings, lubrication troubles, possible loosening of foundation and bearing bolts, calling for frequent adjustment and overhaul, and imposing unnecessary strain on the personnel. (That the condition tending to such results does exist on many of our destroyers I had from very high authority, being informed after the *Shaw's* full-power trials off Santa Barbara, that the *Shaw* was the only turbine destroyer free from critical speeds at which the turbine vibration was excessive.) The above results of dynamic unbalance are readily appreciated. There is still another serious result of dynamic unbalance which is not so apparent and is consequently usually overlooked. This is the loss of power wasted in causing and maintaining vibrations in the structure supporting the rotating body. That there is such a loss of power was conclusively proved by Sommerfield, a German physicist (described in "Zeitschrift des Vereins Deutscher Ingenieure," 1902) in his experiments with a small electric motor carrying an eccentric weight on its shaft. This motor was placed on a table, the circuit closed and the motor

gradually speeded up. It was found that at 310 r.p.m. the supporting table began to vibrate horizontally. An effort was made to increase the motor speed by an increase in the line voltage, but it was found that a considerable increase in voltage did not increase the speed of the motor, the increase of power consumption being wasted in maintaining the vibrations of the supporting table. When this "horsepower expended in producing vibrations" is considered the curve of S.H.P. and knots, U. S. S. *Shaw*,* which was so remarkable as to cast doubt upon its accuracy, may not have been so far off as was supposed at the time. Although designed to make 29½ knots at about 18,625 S.H.P., an examination of the curve shows that 29½ knots was obtained with 580 r.p.m. The r.p.m. and S.H.P. curve shows that 580 r.p.m. corresponds to 15,600 S.H.P

After coming to the above conclusions, it was apparent that one of the essential features of a successful marine turbine installation was a perfect dynamic balance if such could be obtained. Correspondence with some of the shipbuilding yards in the United States elicited the fact that up to that time little attempt had been made to secure dynamic balance, the rotating elements being merely carefully balanced statically. The following is quoted from the General Specifications for machinery for vessels of the U. S. Navy:

"Rotors.—Great care will be taken to insure the rotor being perfectly balanced when complete; first, statically on knife edges; and, second, dynamically under steam in its own casing and at revolutions 25 per cent higher than those specified for full power."

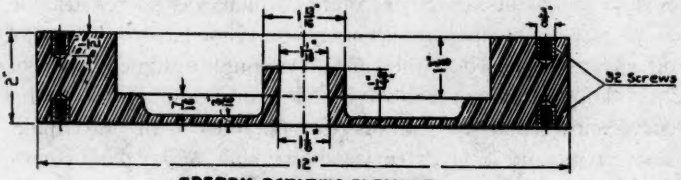
It is apparent that this dynamic test is not a dynamic balance (it will be shown later that, except when dealing with a case of a thin disc, static balance does not necessarily imply dynamic balance; on the contrary, in practically all cases statically balancing any long body will increase its dynamic unbalance).

*The curves of S.H.P. and knots of the *Cassin*, *Cummings* and *McDougal* are as low in proportion, taking into consideration the increased displacement of the *Shaw* and the propellers of these three vessels were not dynamically balanced.—Ed.

I was able to find little or no literature on the subject of dynamic balance of rotating bodies.

(Later the articles on dynamic balance by N. W. Akimoff, published by the A. S. M. E., confirmed many of my assumptions. My only practical experience in dynamic balancing was with the following method used in the dynamic balance of the rotating element of the Sperry gyro compass.)

The rotating element is of the shape and dimensions shown



SPERRY ROTATING ELEMENT.
WEIGHT 47 POUNDS
SPEED 8600 RPM.
FIG. 1.

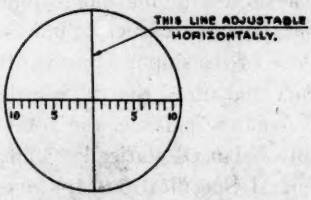


FIG. 2.

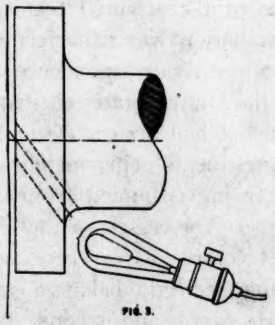


FIG. 3.

in Fig. 1 and weighs 47 pounds. On both sides, near the circumference, holes are drilled and tapped for countersunk screws. After the element is carefully balanced statically it is mounted on a vertical spindle with flexible supports and speeded up to 8,600 r.p.m. The speed of greatest vibration is determined and the element rotated at that speed. A small tube containing a pencil fitting loosely in the tube is approached toward the rotating element. When the pencil point makes contact with the part of the rotor which throws out the maxi-

mum amount, the pencil is knocked back inside the tube, making a mark on the rotor only at the point of contact; the rotor is then rotated in the opposite direction, another mark made, and the heavy point is considered to be midway between the two marks. A shorter screw is substituted for the one inserted at this point and the process repeated until a satisfactory degree of dynamic balance is obtained. Satisfactory results are obtained for the reasons that the rotor is light and easily handled, that it is a small single forging, that it is machined all over and is, consequently, symmetrical and homogeneous, and that as its shape is nearly that of a disc, the static balancing is practically a dynamic balancing as well. It is apparent that this method would not be practicable with a long, heavy rotor made up of several pieces of different diameters and thickness and fitted with thousands of blades of many different lengths which are usually caulked heavier on one side than on the other, making the "density of loading" greater on the side where the heavier caulking is done.

From my study of dynamic balancing I believed that the following assumptions would hold:

(*a*) That in the case of a long rotor, static balancing would in practically all cases increase the dynamic unbalance.

(*b*) That any rotor has one critical speed at which the dynamic unbalance is a maximum, and if dynamically balanced for this speed it will be dynamically balanced for all other speeds.

(*c*) That the quickest and most satisfactory results would follow if the rotor was mounted in bearings that would suppress vibration in the vertical plane while permitting it in the horizontal plane.

(*d*) That the rotor must be so mounted that it can be rotated by an external applied force, but at the same time must be absolutely free from any restraint or impulse from that external force.

(*e*) That the vibration of the rotor must be shown magnified so as to enable it to be more easily observed and corrected.

(*f*) That there must be provided some type of registering device that would indicate not only the point of maximum displacement but also its amount, so that the effect of an added weight (whether it decreased or increased the dynamic unbalance) could be accurately determined.

It was a simple matter to develop a satisfactory design of mounting to provide for assumptions (*c*) and (*d*). Certain features of (*e*) and (*f*) were equally simple.

My first design contemplated a mirror free to oscillate about a vertical axis connected to the vibrating rotor by a wire and an adjustable lever, and a magnifying telescope, having a ground-glass screen with horizontal graduations and with an independent vertical wire adjustable in a horizontal plane (Fig. 2). The telescope to be focused on the image reflected in the mirror of any convenient point on the rotor end. The vibration of the rotor could be magnified to any reasonable amount desired by the adjustable lever on the oscillating mirror and the magnification of the telescope. I originally contemplated drilling a small hole in the rotor shaft (Fig. 3) as shown and placing a fixed powerful light so that it would shine through the hole; when the speed of rotation became sufficiently great, there would appear to be a bright spot in the exact center of the shaft. The reflected image of this spot could be followed with the vertical adjustable wire of the telescope while the speed of the rotor was gradually increased. When the maximum displacement was reached the rotor could be held steady at that speed and the heavy point marked by the use of the Sperry tube and pencil above referred to. I believe that this original device, or some logical simplification of it, would have proved satisfactory, but it was never used, for the reason that about this time I found, in a copy of "London Engineering," a description of a method of dynamic balancing which was practically the same as my design except that there was attached to the end of the rotor shaft a disc with an involute slot through which a light shone and in the telescope the ground-glass screen was graduated differently.

The value of this involute slot was seen at once, as it combined both marking and recording devices. It is possible that I would have independently developed this particular feature later; again, it is possible that I would not. Full acknowledgment of its value is made, however, as a great amount of time, trouble and experimenting was saved. From a mechanical standpoint I found nothing worthy of adoption in the method described in "London Engineering." On the contrary, some of the features described were distinctly inferior to our own design.

II.—GENERAL DISCUSSION OF DYNAMIC BALANCING.

Any rigid rotating body can be regarded as being made up of a very great number of thin elements or cross sections, each of which tends to rotate about the axis passing through its individual center of gravity. If the centers of gravity of all the elements lie in the axis about which the body rotates, the body is in dynamic balance and will run quietly and without vibration at all speeds. If, however, the centers of gravity of some of the elements do not lie in the axis of the body, these elements will each tend to rotate about its individual center of gravity, setting up unbalanced forces, and the body as a whole will be in dynamic unbalance, and may be in static unbalance as well, resulting in vibrations and excessive twisting and bending stresses in the shaft and unnecessary friction, heat and wear of the journals.

Fig. 4 represents a rotor which would be in perfect balance both statically and dynamically if it were not for the existence of a single symmetrical eccentric mass of weight A_1 pounds. If r_1 is the distance of the center of A_1 from the center of the rotor and x and y are the distances respectively from the center of the mass A_1 to the centers of the journals, it will be apparent that if the rotor is rotating with an angular velocity of ω, the journal bearings J and J_1 will each be subjected to a force in pounds as follows:

$$\frac{A_1\omega^2 r_1}{g} \times \frac{y}{x+y} \text{ and } \frac{A_1\omega^2 r_1}{g} \times \frac{x}{x+y}$$

These forces are reversed twice for each revolution of the rotor. It is evident that with high speeds of rotation a small unbalanced mass will set up excessive vibrations at the journal bearings.

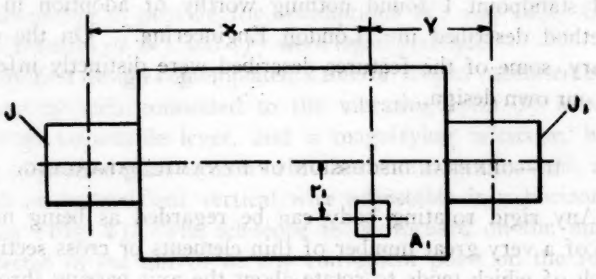


FIG. 4.

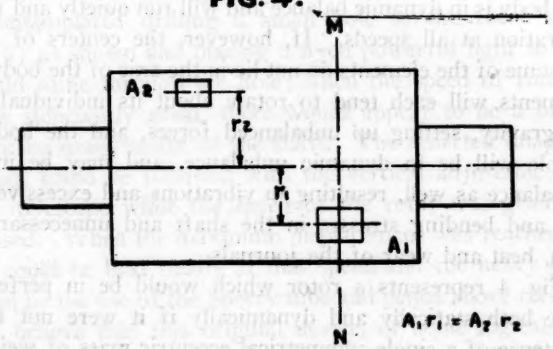


FIG. 5

If it was desired to statically balance the rotor shown in Fig. 4, it would be placed on knife edges, the heavy spot located and at 180 degrees from A_1 a weight, A_2 (Fig. 5) so placed as to perfectly counterbalance the mass A_1.

This applied weight, A_2, might be greater or less than the mass A_1, and it might be placed at a distance r_2 from the axis greater or less than r_1. If $A_1 r_1 = A_2 r_2$, perfect static balance will be obtained, but it is apparent that unless A_2 was by chance located in the same plane MN, as A_1, the dynamic unbalance would be increased, as in addition to the forces due to A_1 acting on the journals and bearings there would be added the forces due to A_2. With a solid rotor of any length there is not one chance in one thousand that the applied weight will be placed in the same plane as the unbalanced mass. The unbalanced mass A_1 may be irregular in form, or there may be several unbalanced masses. Again, in the case of a heavy rotor, to avoid injury to the journals, the knife edges must be somewhat blunt, which renders the static balance on knife edges so insensitive that a relatively heavy weight can be applied without effect. These considerations show the hopelessness of attempting to secure dynamic balance by merely statically balancing the rotor.

III.—DESCRIPTION OF APPARATUS, METHOD OF OPERATION, AND RESULTS OBTAINED IN BALANCING THE CRUISING, H.P., L.P. AND ASTERN, AND STARBOARD ASTERN ROTORS, REDUCTION GEAR AND TWO PROPELLERS, U. S. S. "SHAW."

The attached photostat, Plate I, shows the H.P. rotor mounted on the balancing stand and connected to the indicating and recording apparatus. This rotor weighed 14,500 pounds. The forging for the drum was received rough-turned and bored. It was concentrically finish-turned and bored, making the wall thickness the same all around the circumference, so that the drum can be considered in perfect static balance. The two end spiders and shaft were received rough-turned; they were machined out between the spokes, finish machined and carefully balanced statically, the drum and end spiders were shrunk together and also secured by heavy machine screws riveted over on both ends. The blading grooves were cut, the blades inserted and caulked, and the binding strips soldered to the blades.

The rotor journals, J_1, J_2, rest in babbitt-lined bearing boxes, B_1, B_2, the outside of the boxes being spherical in shape. These boxes are fitted into movable female spherical bearings, providing a ball joint, so that the rotor is free to oscillate without binding. These movable bearings, A_1, A_2, rest on rollers. Two holding-down bolts pass through elongated holes in each end of the bearing. The holding-down bolts are free—their purpose being to prevent the rotor from jumping off the stand in case of excessive vibration. The bearings are free to move laterally, excessive lateral movement being prevented by the four restraining springs, S_1, S_2. These springs were ground to the same exact length and were accurately calibrated by grinding the outside of the coil as necessary until each spring was compressed the same distance by the same weight. The tension of these springs can be adjusted as desired by the threaded bolts, F_1, F_2, passing through the transoms of the fixed bearings, C_1, C_2, and bearing against washers fitted to the springs. The end of the shaft of the driving motor is fitted with a flange having four projecting pins, the end of the rotor shaft being similarly fitted. A leather driving belt is passed around these pins as shown, Plate II, so that while the motor rotates the rotor, the rotor is not restrained laterally but is free to vibrate if in dynamic unbalance.

At the opposite end of the rotor is fitted an aluminum disc with a forged-steel rim shrunk on for strength. This disc has an involute slot cut through (the origin of the involute slot is at the center of the shaft; the slot itself, however, begins just outside of the hub and continues through a little more than a circumference, bridges being left at certain points for strength and stiffness). The form of the slot is an involute, for the reason that equal angles of rotation correspond with equal radial displacement from the center toward the circumference. This disc is marked with four radial lines 90 degrees apart, marked 1, 2, 3, 4, in line with similar marks on the rotor. A light box with a long vertical slot is placed behind the disc. This box contains two long-filament lamps. A

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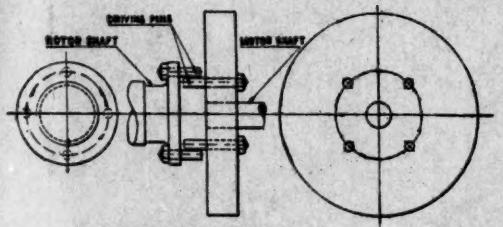
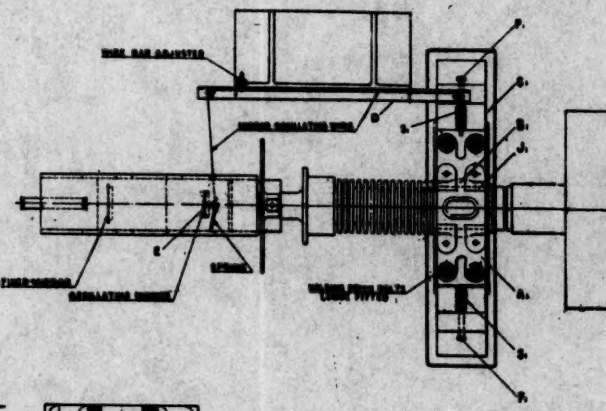
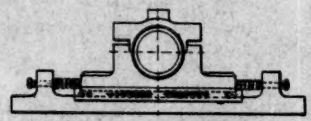


FLEXIBLE COUPLING
ROWING BANDS REMOVED.

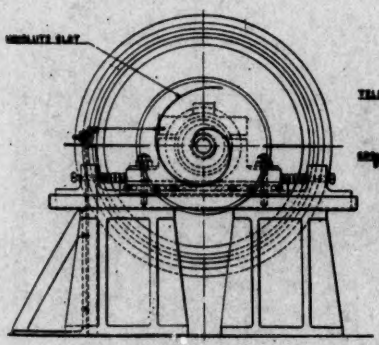


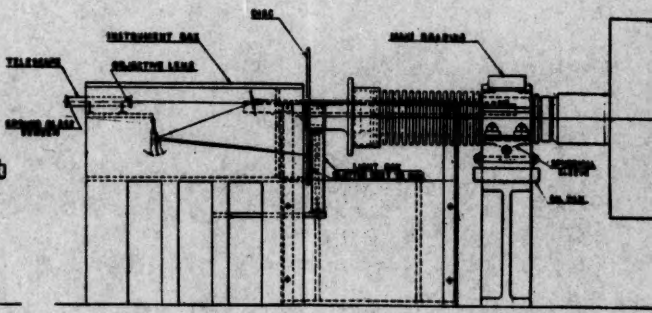
MALE AND FEMALE SPHERICAL BEARING.

POD ROTORS POD PROPELLERS
GROUND GLASS SCREENS.
½ SIZE



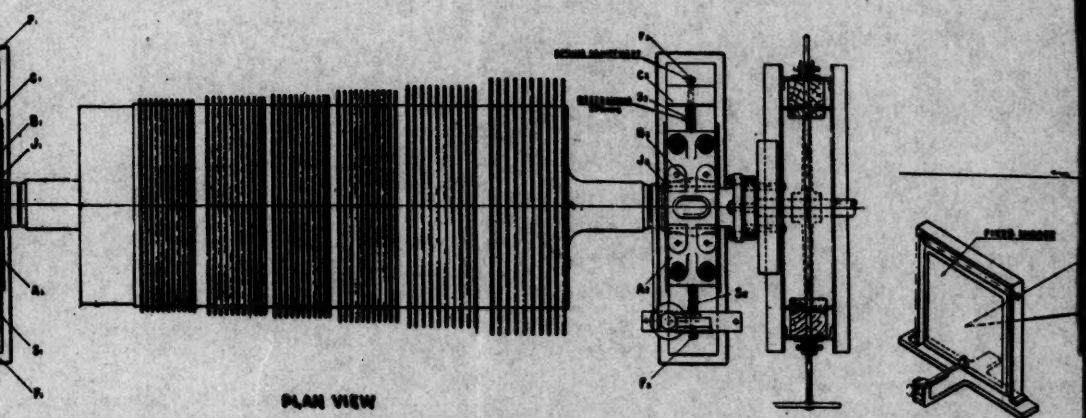
MOVABLE AND FIXED BEARING.





FRONT END
INSTRUMENT BOX REMOVED.

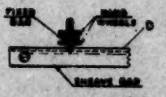
H.R ROTOR, U



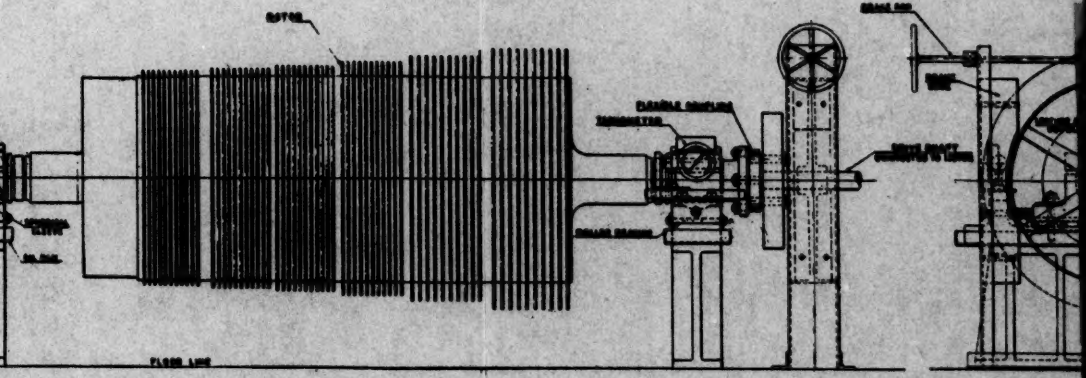
PLAN VIEW

DYNAMIC BALANCING MACHINE

FOR PROPELLER SCREENS.



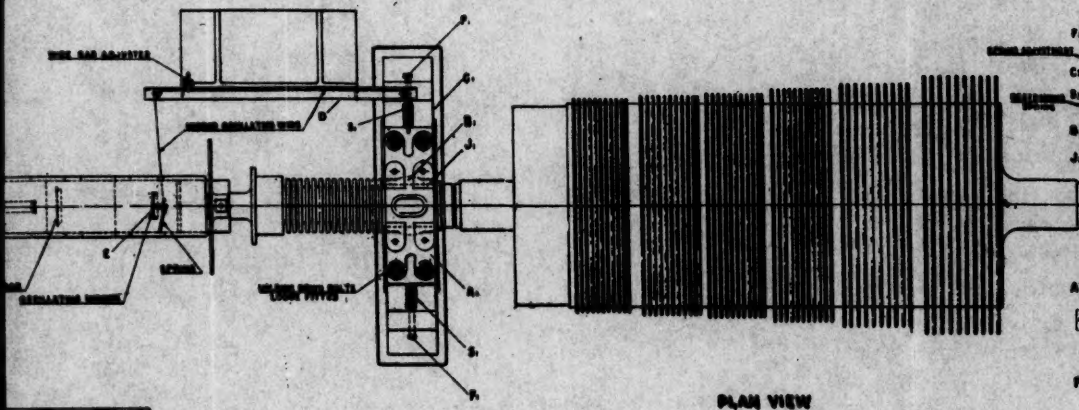
WIRE BAR ADJUSTER



SIDE ELEVATION

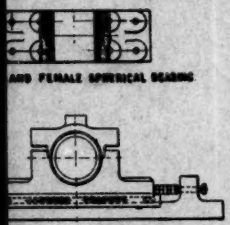
H.R. ROTOR, U.S.S. SHAW, IN BALANCING POSITION.

PLATE I.



PLAN VIEW

DYNAMIC BALANCING MACHINE

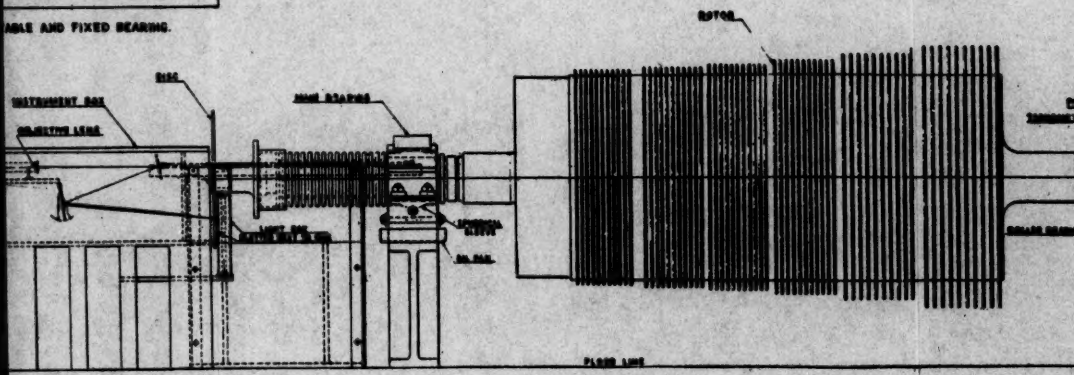


AND FEMALE SPHERICAL BEARING.

FOR ROTORS
GROUND GLASS
½ SIZE

FOR PROPELLERS
SCREENS.

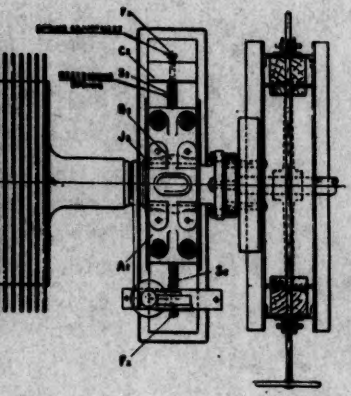
ABLE AND FIXED BEARING.

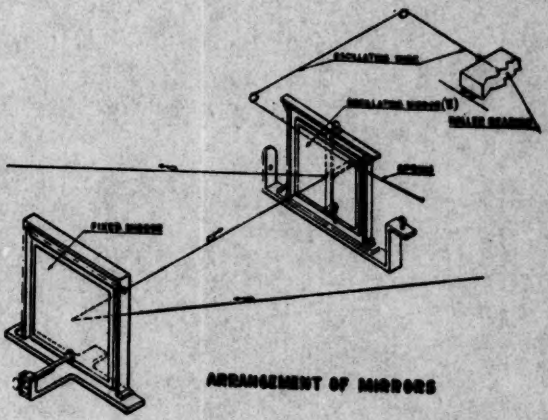


SIDE ELEVATION

H.R ROTOR, U.S.S. SHAW, IN BALANCING POSITION.

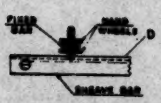
PLATE I.



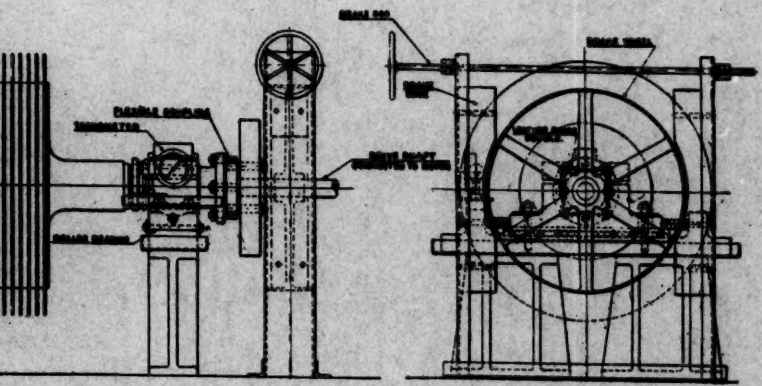


ARRANGEMENT OF MIRRORS

INE



WIRE BAR ADJUSTER



BACK END

PLATE II.

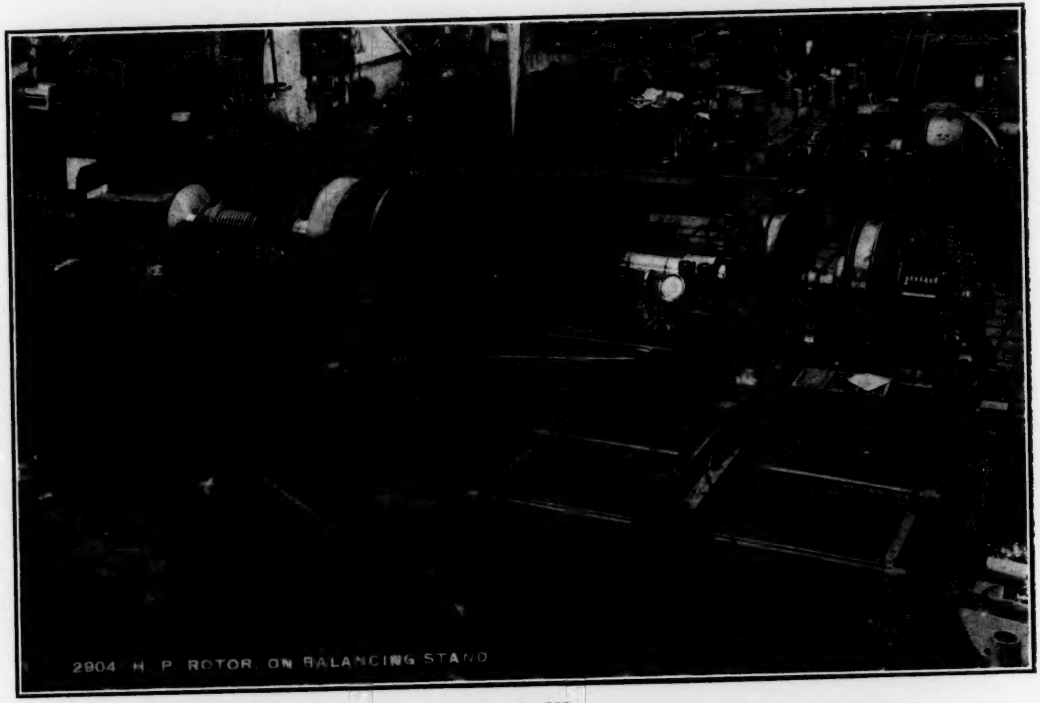


PLATE III.

PLATE IV.

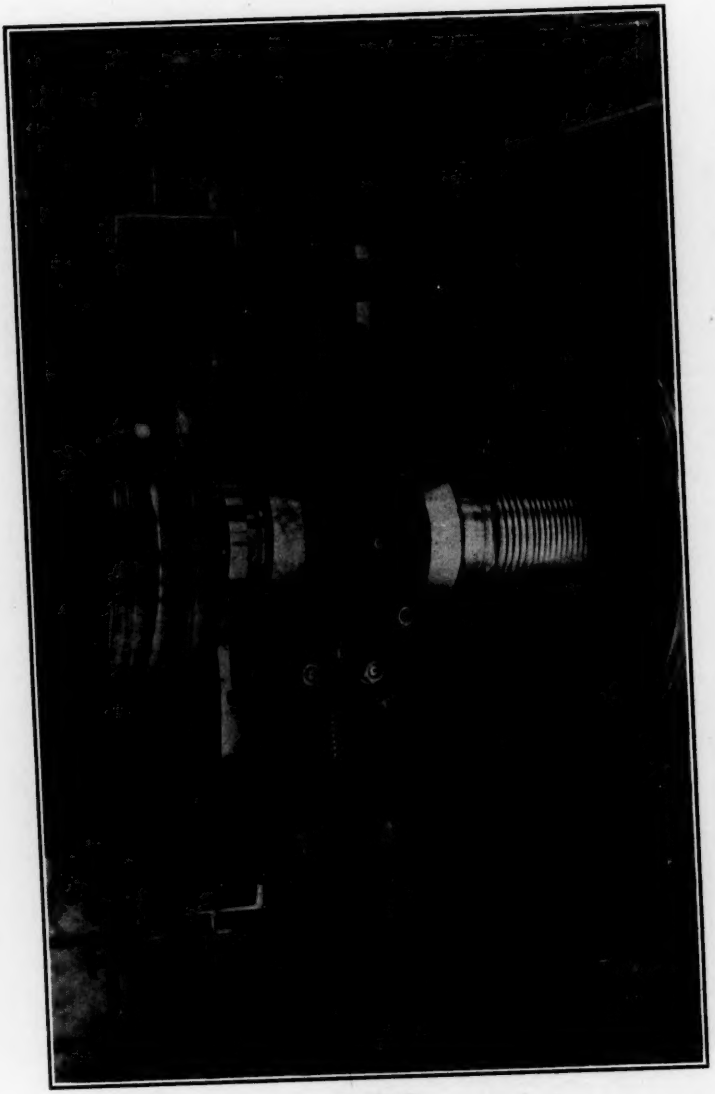


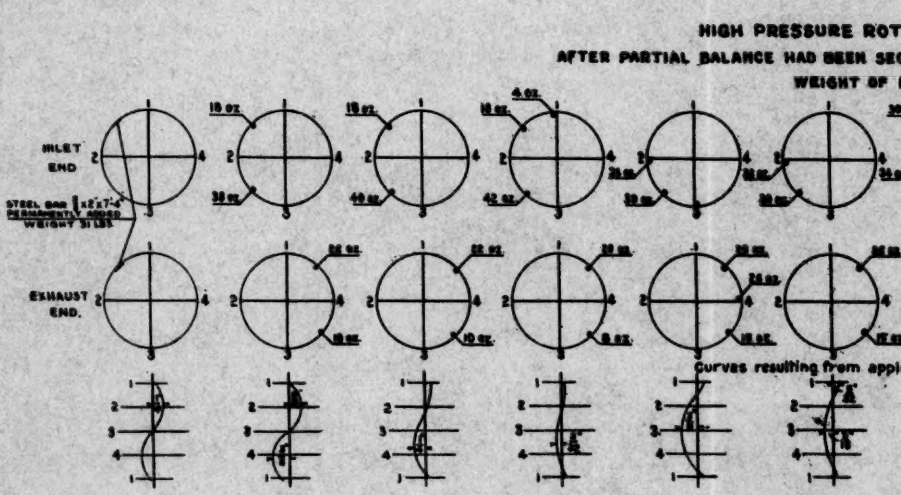
PLATE V.

PLATE VI.

HIGH PRESSURE ROT

AFTER PARTIAL BALANCE HAD BEEN SE

WEIGHT OF



Curves resulting from appl

450 RPM

CRITICAL SPEE

SCALE

BOTH ENDS LOOKING TO

FI

After the weights were se
was noticed on both ends.
were added to the weight
1 and 2 and 4 oz. more adde
the inlet end. On the exhau
shifted to 3 as shown bel

PLATE V

ESSURE ROTOR NOV. 3-1916.

HAD BEEN SECURED BY ADDING 31 LB. STEEL BAR.

WEIGHT OF ROTOR 14500 LBS.

ulting from applied weights shown just below.

450 RPM 480 RPM 400 RPM 650 RPM 480 RPM

CRITICAL SPEED VARIABLE.

SCALE $\frac{1}{2}$ SIZE

THE ABOVE WEIGHTS WERE SECURED IN POSITIONS SHOWN.

NDS LOOKING TOWARD EXHAUST END:

FINAL

eights were secured a slight vibration on both ends. To overcome this, 4 oz. to the weight on the spoke between 4oz. more added between 2 and 3 on d. On the exhaust end the 3 oz on 1 were 3 as shown below.

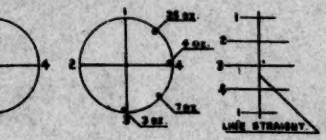


PLATE VII.

mirror is fitted to the inside of the box opposite the slot so that all the light possible will pass out through the slot. The light box is not connected in any way with the rotor. To the cap of one of the movable bearings a wire is secured and passes around two sheaves on an adjustable sheave bar, D, the other end of the wire being secured to a lever on the back of the oscillating mirror, E. A spring is attached to the lever on the side opposite to the wire so that the wire moves the mirror against the tension of the spring. The wire is thus held taut at all times. Vibration of the rotor oscillates the mirror. The sheave bar, D, carrying the two sheaves, is adjustable by means of two hand wheels so that when the rotor is at rest the mirror may be set in the central position. The wire and spring may be shifted to different holes in the lever, varying the leverage and thus increasing or decreasing the amount of oscillation of the mirror for the same amount of vibration of the rotor.

The light shining through the vertical slot in the light box is shut off by the disc except at the point where the involute slot crosses the vertical slot, creating a luminous point. A ray of light from this point strikes the fixed mirror, is reflected from the fixed mirror to the oscillating mirror, then reflected into the telescope, and is brought to a focus on the ground-glass screen. The fixed mirror is adjustable about a horizontal axis by means of a thumbscrew for the purpose of moving the point of light in a vertical direction on the ground-glass screen to set the point of light with the rotor. In adjusting for a run the rotor is jacked over until the quadrant line 1 on the rotor is in the horizontal plane, determined by a scriber from the floor plate. The hand wheels on the wire bar adjuster are moved until the point of light is on the vertical line on the ground-glass screen, the thumbscrew on the fixed mirror is then moved until the point of light is on the intersection of the vertical line and the horizontal line 1 on the ground-glass screen. Quadrant lines 2, 3 and 4 are checked in the same way.

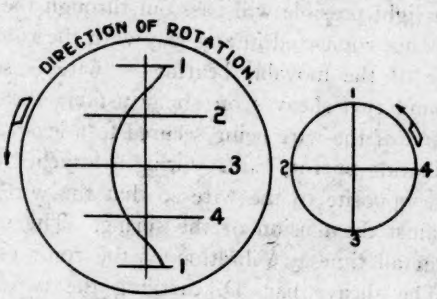


FIG 6..

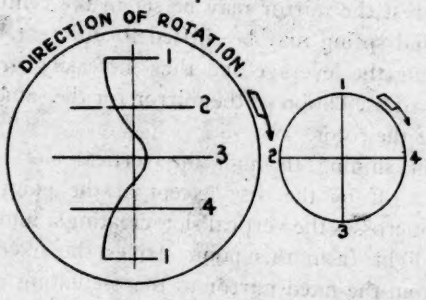


FIG 7

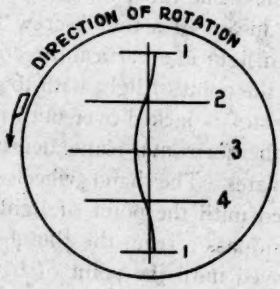


FIG. 8.

The motor is then started and the speed of the rotor slowly increased; the point of light travels slowly up and down the vertical line on the screen; when the speed of rotation becomes sufficiently high, the moving point of light appears as a continuous bright line. This line of light remains on the vertical line on the screen until the lowest oscillating speed of the rotor is reached and the rotor begins to vibrate. The vibration of the rotor moves the oscillating mirror and the hitherto vertical line of light becomes curved on both sides of the vertical center line on the screen.

The ground-glass screen is marked as shown, the horizontal lines marked 1, 2, 3, 4, 1 correspond to the similarly marked radial lines on the rotor and on the involute disc.

The speed of the rotor is gradually increased until the maximum speed at which the rotor will operate in active service is reached, the movement of the line on the ground-glass screen is carefully watched, and note is made of the speed of the rotor at which the line has the greatest lateral displacement. This point of greatest displacement or critical point seldom corresponds with the maximum rotor speed. The rotor is then steadied at the critical speed and a sketch made showing the appearance of the line on the ground-glass screen (Fig. 6).

An examination of this line shows that the maximum displacement occurs at 3, indicating that the heavy spot on the rotor is located on the radial line 3. This would be exact were it not for the inertia due to friction on the journals which causes a lag of the effect behind the cause. In other words, the heavy spot is somewhat in advance (in the direction of rotation) of the point showing the maximum displacement on the screen. This lag makes it necessary to rotate the rotor in the opposite direction *at the same critical speed,* giving the record shown in Fig. 7. In Fig. 6, the heavy spot on the rotor is in advance (counterclockwise) of 3. In Fig. 7, the heavy spot on the rotor is in advance (clockwise) of 1, consequently the heavy spot is midway between 3 and 1, or at 4. A weight

of a few ounces is secured on the rotor at 2, as near the periphery as possible, and the rotor rotated in either direction up to the critical speed, and while steady at that speed another record is made (Fig. 8). A comparison of this record with Fig. 6 (same direction of rotation) will clearly indicate whether the dynamic unbalance has been decreased or increased. The weight at 2 should be increased and decreased until the displacement is a minimum. Record it by a sketch, then shift the weight to 4 and follow the same procedure. When both the weight and the location causing the minimum displacement is determined, secure the weight at that point. Follow the same procedure at the other end of the rotor. (It may be that any weight at the other end increases the dynamic unbalance; if so, no weight should be applied at the other end.)

Now rotate the rotor up to the maximum designed speed, both clockwise and counterclockwise. A new critical speed and a new heavy spot may be obtained. The new heavy spot will be less in magnitude and is to be dealt with in the way already described. The number and location of all the balance weights can be determined by successive trials. In the case of the cruising rotor of the *Shaw*, perfect dynamic balance was obtained by the application of one weight. This was also the case with the reduction gear. The starboard astern rotor, the high-pressure rotor, and the low-pressure and astern rotor required several weights at each end.

The balancing of the first rotor (cruising) was somewhat tedious, owing to the fact that the method was new to all of us. Many minor changes had to be made, considerable experimenting done, and each successive result obtained had to be carefully analyzed. The time spent and the experiments made on this first rotor were well repaid by the results obtained. The employees who mainly assisted me—Leadingman Gordon and Apprentice Neilson—so readily acquired the knowledge necessary for the work that they were able to do the greater part of the work of balancing the other units, as I confined myself to supervising their work and checking their results at frequent intervals.

The rotors, reduction gear and propellers were balanced by placing weights in various places, as previously described. In the case of the starboard astern rotor, high-pressure rotor and low-pressure and astern rotor, it was an easy matter to secure equivalent weights in the indicated places by countersunk screws riveted over.

The cruising rotor, reduction gear and propellers were not so easily handled. In the case of the cruising rotor, a hole was drilled at the same radial distance and diametrically opposite to the applied balance weight, the drillings carefully weighed and a threaded plug of the same weight inserted. The weight of this plug was reduced about ½ ounce at a time by turning the surface at a point from the end of the rotor equal to the distance of the balance weight from the same end. The balance weight was then reduced an equal amount and the rotor run again. In this way, a constant check was kept, and when the last balance weight was finally entirely removed the threaded plug needed very little additional machining. The end of the plug was then riveted over.

In the case of the reduction gear, the balance was very quickly secured by adding weights. The added weights were found to be exactly equal in weight to one of the lock nuts on one of the through bolts. The added weights were removed, also the lock nut, 180 degrees away, after which a check run showed that the gear was still in perfect dynamic balance. The through bolt was then turned down in the center until an amount equal to the weight of the lock nut was removed and nut then replaced.

In the case of the propellers, after they were planed on the driving (after) side, three threaded steel studs were screwed into the forward face of each blade on the same circumference, spaced as shown (A, A, A, Fig. 9). The applied balance weights were in the shape of various sized washers secured by a single nut. As all tap bolts weighed exactly the same and were symmetrically placed, their weight was neglected and only the balance weights considered. Owing to the fact that

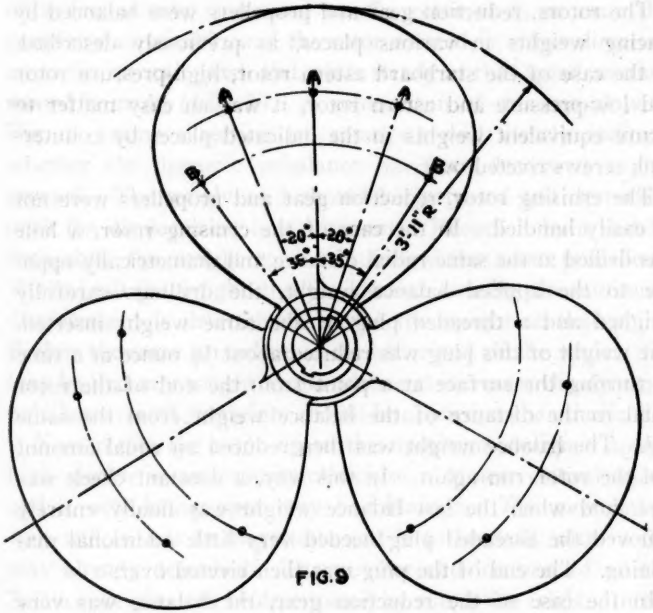


FIG. 9

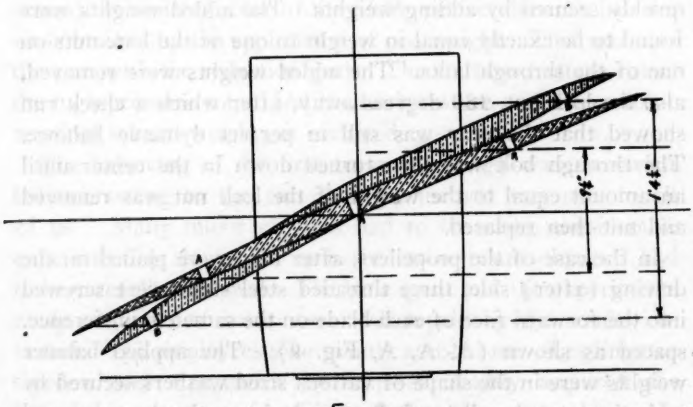


Fig. 10

the hub of the propeller approximated in shape to the frustum of a cone it was found that the three tap bolts as originally placed were too nearly in an athwartship line to obtain satis-

factory results (A, A, A, Fig. 10). Consequently the center tap bolt was left in its original position, the two side bolts being shifted closer to the hub (B, B, Fig. 9). This shift moved one of the bolts farther forward and the other farther aft (B, B, Fig. 10). It was not at all difficult to balance the propellers by applying weights. As it is apparent that these weights could not remain on the propeller, it was necessary to chip and file metal off the blades opposite each weight, and as the blades were comparatively thin near the edges where the weights were applied, the chipping had to be done nearer the hub, and consequently the weight of metal chipped off was greater than the applied balance weight opposite. The procedure followed was to remove one of the balance weights, weigh it, and then chip off metal as nearly diametrically opposite as could be done. Then another trial run would be made, another balance weight removed and more metal chipped off. This process was continued until the propeller was in perfect dynamic balance with all balance weights removed. The steel studs were then removed and the tap holes filled with bronze studs cut off and riveted over.

It is of interest to note that while the first (starboard) propeller was statically balanced on knife edges, more than thirty pounds of metal being chipped off the blades, nevertheless, it was necessary to apply 22¾ pounds in weights to dynamically balance it; in several cases these weights were applied at about the same points where metal had been chipped off. Consequently, no attempt was made to statically balance the second (port) propeller, and it was observed that on its first runs it was apparently as well balanced dynamically as the first propeller which had been put in static balance. Thirty-seven and one-fourth pounds was applied in balance weights to dynamically balance this propeller.*

During the process of balancing the rotors the question arose

*The accuracy of the experiments and deductions are not questioned, but it is difficult to realize how a statically balanced propeller can be so far out of dynamic balance when the short horizontal length of the propeller is considered.—Ed.

whether the condition of the rotors being driven by motor against the wind resistance would not be so different from the same rotors under steam driving the shaft and propeller that the dynamic balance attained under the former condition might not hold under the latter. Also, as to whether the condition of the propellers being balanced in air with equal resistance and pressure all over might not be entirely different from the same propellers revolving in water with the tip of the upper blade operating under entirely different conditions from the tip of the lower blade. Great interest was therefore taken when the turbines were run free under steam and also during the *Shaw's* trial runs. All turbines were run for two hours under steam, using 160 pounds steam from the boilers of the central power plant. All turbines ran without any appreciable vibration. The high-pressure turbine calls for special comment. The feet of the casing were clamped to a 10-foot × 5-foot × 6-inch bed plate, and this in turn was clamped to the rails of the track running into the basement of the central power plant. After the rotor and casing had been warmed up and the rotor turned over slowly and then stopped, I removed the rubber eraser from a lead pencil 6 inches in length and 5/16 inch in diameter and stood it upright on the nut of one of the flange bolts. The steam valve was then slowly opened and the rotor was gradually brought up to its maximum speed and maintained at that speed for ten minutes, the steam being then shut off and the rotor allowed to come to rest. During this entire time of 25 minutes there was such entire absence of vibration that the pencil maintained its upright position throughout. The dock trials, shaking-down cruise, full-power runs and run from Mare Island to the East coast, together with the report of her commanding officer, are sufficient proof of the behavior of the turbines and propellers while steaming under actual service conditions. That the dynamic balancing of the rotating elements of the *Shaw's* turbines contributed materially to her efficiency is unquestionable. All of the assembled rotating elements of the tur-

bines were tested for static balance and found satisfactory in that respect. Yet every element, when mounted on the balancing stand and rotated, vibrated excessively. In the case of the two propellers and of the heavy high-pressure and low-pressure and astern rotors, the vibration of the rotor and the stand was so violent as to be dangerous at comparatively low speeds of rotation, and the propellers and rotors had to be partially balanced dynamically before they could be safely rotated up to the higher speeds. The completely finished high-pressure rotor was tested for static balance by rolling on two parallel rails at the journals and no lack of static balance could be detected. Yet it required a flat steel bar 7 feet 4 inches × 2 inches × ⅝ inch, weighing 31 pounds, and other weights totalling 9 pounds more, in all 40 pounds, added at various points to secure perfect dynamic balance. This rotor weighed 14,500 pounds and, when finally balanced, ran at all speeds on the balancing stand without any appreciable vibration. Yet to illustrate the importance of dynamic balancing and the sensitiveness of the balancing device, it is sufficient to state that when one of the balance weights, weighing 30 ounces, was removed as a test, at 220 r.p.m. the vibrations of the rotor were plainly noticeable to the eye and could be felt by the hand placed on any part of the balancing stand, and the line of light on the ground-glass screen was displaced ⅛ inch on both sides of the vertical center line.

There is one very striking advantage which this optical indicating and recording device possesses over other balancing methods; that is, the magnification of the amount of vibration of the rotor at the bearings. This magnification enables the observer to much more readily determine the heavy spots and to make a far more accurate adjustment. The amount of this magnification or optical leverage is determined by the following formula (Fig. 11):

$\frac{2Ll_1}{l_1 + l_2} : l$, where the oscillating wire is connected to the lever of the oscillating mirror at a point 1 inch from the pivot.

Where

L, is the distance from the telescope objective lens to the ground-glass screen;

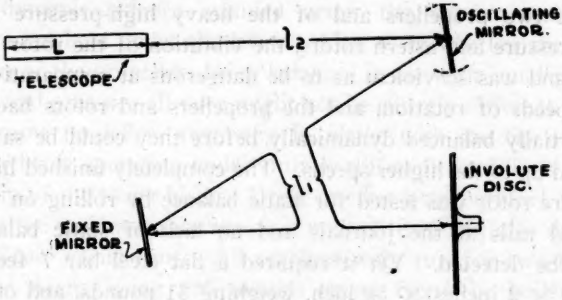


FIG. II.

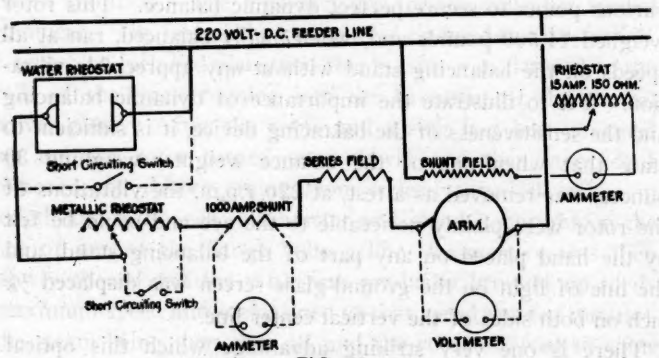


FIG 12.

l_1 is the distance from the involute disc to the oscillating mirror;

l_2 is the distance from the oscillating mirror to the telescope objective lens.

By selection of a lens of the proper focal length (which determines L) and by suitably placing the mirrors and telescope, the optical leverage can be made any value desired.

In my work here, only one telescope lens was available, which gave L = 15 inches. Preliminary experimental work

determined the most suitable radial distance from inner to outer end of the involute slot to be 11⅜ inches; as the telescope had a field of 12⅞ inches at 84 inches from the objective lens, l_1 and l_2 were respectively taken as 60 inches and 24 inches, giving an optical leverage of

$$\frac{2Ll_1}{l_1 + l_2} : 1 = \frac{2 \times 15 \times 60}{60 + 24} = \frac{150}{7} = 21.$$

To provide for varying this optical leverage three holes were drilled in the lever arm of the oscillating mirror at distances from the pivot of 1 inch, 2 inches and 3 inches. Attaching the oscillating wire to these different holes gave optical leverages of 21, 10.5 and 7.

Owing to inaccuracies in making the instrument box, placing the mirrors, etc., actual test showed the optical leverages to be 12, 6 and 4.

The lower optical leverages were used in the first stages of balancing, as the use of the maximum leverage resulted in the line of light being thrown completely off the screen. The optical leverage of 12 was used after the vibrations had been reduced by partial balancing.

In the case of the high-pressure rotor referred to on page 19, where a 30-ounce weight was removed, the actual vibration of the rotor was ⅛ inch $\times$ 1/12 = 1/96 inch each side of the mid position; yet this small amount of vibration, as already stated, was plainly noticeable to the eye and could be felt when the hand was placed on any part of the balancing stand.

The power-driving facilities were satisfactory for the cruising and starboard astern rotors and the reduction gear. They were not adequate to drive the high-pressure and low-pressure and astern rotors against the wind resistance at the higher speeds.

It is necessary to use a d.c. motor on account of the nice control of speed required. The only suitable location for balancing was the erecting floor in Shop 31. Owing to the distance from the central power plant and the small size of the d.c. feeders the voltage drop was considerable. The voltage

input in the lines at the power plant was 280, the voltage at the motor was 165; voltage drop, 115, or 41 per cent. It was necessary to do most of the balancing of these two rotors after the close of the working day so as not to interfere with the operation of the overhead (d.c.) cranes.

Many makeshifts were necessary, and even then it was possible to bring the high-pressure rotor up to 670 r.p.m. (maximum r.p.m. for full-power plus 25 per cent.) only when running in the ahead direction when the convex side of the blades was presented to the wind resistance. When the concave side of the blades was presented to the wind resistance the maximum number of r.p.m. attainable was 600. But this speed was in excess of the critical speed. With the low-pressure and astern rotor the r.p.m. attained were even lower; the maximum ahead speed obtained was 560 and the maximum astern speed was 425. In the case of the two propellers, the maximum r.p.m. attained were 425 both ahead and astern. Fortunately the speed of maximum vibration occurred in all cases within the speeds attainable. The attached wiring diagram, Fig. 12, shows clearly the electrical arrangement for the low-pressure and astern rotor drive. The maximum work required of this arrangement was driving the low-pressure and astern rotor at 560 r.p.m. A compound-wound generator of 100 k.w. rated output was used as a motor on the d.c. feeders and developed 210 H.P., using 165 volts d.c. at 950 ampères.

TESTS OF MAIN CIRCULATING PUMPS OF *NORTH DAKOTA* AND *NEVADA*.

By Commander S. M. Robinson, U. S. Navy, Member.

The test of the *Nevada's* main circulating pump was made by Commander O. L. Cox, Senior Engineer Officer of that vessel, and all of the test was carried out on board ship. The test of the *North Dakota's* circulating pump was made by the Fore River Shipbuilding Company, and part of the test was carried out ashore and the remainder of the test was carried out on the ship.

On the *Nevada* the static head was measured at four points, as shown in figure 3. No. 1 was in the main injection pipe; No. 2 was at the entrance to the main condenser; No. 3 was at the middle point of the rear water chest of the main condenser; and No. 4 was at the outlet from the main condenser. Point No. 1 was 4 feet 3 inches above the base line of the ship; point No. 2 was 13 feet 10 inches above point No. 1; point No. 3 was 15 feet ½ inch above No. 1; point No. 4 was 18 feet 11 inches above No. 1. The water line was 26 feet 4 inches above point No. 1, or 30 feet 7 inches above the base line.

A ½-inch pipe was led from each of these points to a manifold under the third deck, as shown in figure 3. Each of these pipes was provided with a valve for shutting it off from the manifold. From this manifold a single ½-inch pipe was led up the engine-room hatch, terminating in a flexible hose, which was attached to a 10-foot gage board, so arranged that the gage board could be raised or lowered, depending on the position of the water in it. All readings given in the table were measured from point No. 1.

The reading at each of the four points for each speed of pump was obtained by opening the valve connecting that

point to the manifold (with the three other valves closed) and then obtaining the reading on the 10-foot gage board. The valve to this point was then closed and that to the next one opened, until a complete set of readings was obtained. Each set of readings for each speed was checked until the readings agreed within less than 1 inch.

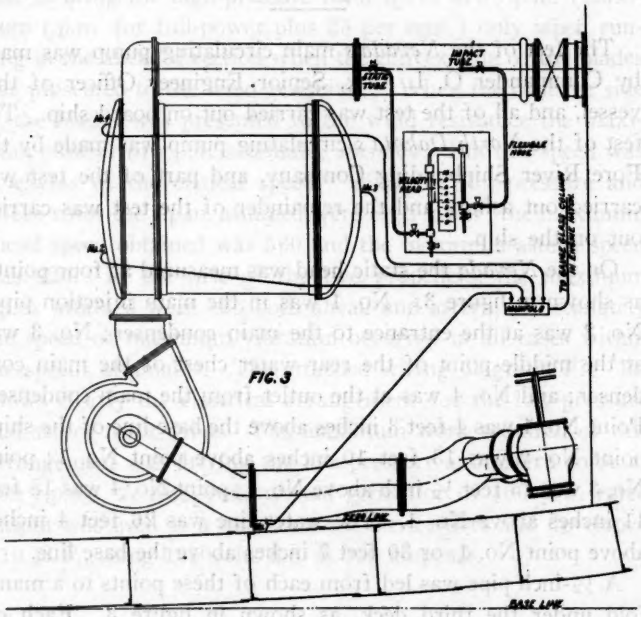


FIG. 3

The velocity head was measured in the overboard discharge pipe by means of pitot tube connected as shown in figure 3; the impact tube was placed at .7 of the radius of the pipe from the bottom of the pipe. This tube was connected to one leg of a "U" tube and a static connection from the overboard-discharge pipe was made to the other leg of the "U" tube. This "U" tube was a closed circuit and was provided with valves for shutting off both connections from it and also with a valve at the top for venting off enough air to bring the readings of

the tube on the gage board, which was placed between the two legs of the tube. The proper position of the impact tube was verified by taking a set of readings over the entire diameter of the pipe, and the average head was found to agree with that obtained by a single reading with the impact tube placed at .7 of the pipe radius from the circumference of the pipe.

Two series of runs were made, the first run being made with the condenser drained and the second under service conditions.

The data obtained are recorded in Table 1. Columns 1, 2, 3, 4, 6, 7, 8 and 9 give the data actually obtained on the test. Columns 5 and 10 give the data derived from that obtained on the trial. The capacities given in column 5 were obtained from the formula—Capacity in gallons per minute= $\frac{8\sqrt{12h}\times 660.5\times 60,}{231}$ where h=velocity head in inches of water and 660.5=area of overboard-discharge pipe, which is 29 inches in diameter. The total heads given in column 10 were obtained by taking the difference between the readings of point No. 1 and point No. 2, given in columns 6 and 7. This difference will give the total head, except the small head due to the friction of the short amount of piping between these two points; this amount is so small that it can be entirely neglected as having no effect on the calculations; the velocity heads cancel each other, since the suction and discharge pipes are the same size. The data given in columns 3, 5 and 10 are plotted in figure 1. Fair curves were drawn, and it will be noted that the points are all quite close to the curves, with the exception of one point on the capacity line. The data obtained from the smooth curves in figure 1 are recorded in Table 2. Columns 1, 2, 3 and 4 give the data taken from the curves. Columns 5 and 6 give the data derived from that taken from the curves. The water horsepower, given in column 5, was obtained from the formula—Water horsepower=

$$\frac{\text{total head (in feet)}\times\text{Capacity (in gallons per minute)}}{3{,}870};$$

the constant used was 3,870 instead of 3,960, as the water was

salt. The pump efficiencies given in column 6 were obtained from the formula—Pump efficiency $= \frac{\text{water horsepower}}{.92 \times \text{indicated horsepower}}$; the constant of .92 was used, as the engine was supplied with forced lubrication; no attempt was made to use the "no load I.H.P." given in column 2, Table 1, for the purpose of obtaining shaft horsepower, as the powers measured were too small to admit of accuracy. The data given in columns 5 and 6 of Table 2 are plotted in figure 1. From an inspection of figure 1, it will be seen that the shape of the various curves follows the laws usually attributed to circulating pumps working on condensers. The capacity curve is a straight line, the capacity varying directly as the revolutions; the total head curve varies approximately as the square of the revolutions; the horsepower curves vary approximately as the cube of the revolutions. The efficiency curve is fairly flat over a wide range, from 9,000 gallons per minute capacity up to 20,500 gallons per minute capacity; it will be noted that this curve is not smooth, but has a change of curvature, and the same phenomenon will be observed in the efficiency curve of the *North Dakota's* pump. This curve, and also the head curves, will be discussed again after taking up the test of the *North Dakota's* pump.

The data on the test of the *North Dakota's* pump were obtained as follows: Two complete sets of tests were run in the shop, the capacity of the pump being measured by a weir; each of these two tests was run with varying revolutions and capacities, and practically a constant head; the total head in one set of tests was about 14.5 feet and in the other about 25 feet.

TABLE 1

1	2	3	4	5	6	7	8	9	10
R.P.M.	I.H.P. (NO LOAD)	I.H.P. (LOADED)	h VELOCITY HEAD IN INCHES OF WATER	Q CAPACITY IN GALLONS PER MINUTE = $\frac{A \times 144 \times 60 \times 62.5 \times 0.6}{231}$	#1 READING OF WATER COLUMN AT POINT NO.1	#2 READING OF WATER COLUMN AT POINT No. 2	#3 READING OF WATER COLUMN AT POINT No. 3	#4 READING OF WATER COLUMN AT POINT No. 4	H TOTAL HEAD IN FEET OF WATER = Col. 7 - Col. 6
75	—	8.0	1.5	5765	25.91	28.53	28.10	26.75	2.62
95	7.5	18.0	2.875	8080	25.57	29.50	28.46	27.68	4.13
115	9.5	22.2	4.0	9525	24.70	30.73	29.10	28.06	6.03
135	11.5	35.5	5.0	10650	24.20	32.30	29.80	28.60	8.10
155	13.75	52.0	7.875	13360	23.90	34.23	30.70	28.80	10.33
175	16.25	72.2	9.25	14480	23.90	36.78	31.60	30.00	12.88
185	18.0	86.2	10.5	15430	23.10	39.48	32.85	30.45	16.38
215	20.0	125.0	13.0	17170	21.90	42.33	34.60	30.65	20.43
235	21.0	160.4	16.0	19050	20.70	45.03	36.00	30.66	24.33
255	22.0	200.5	18.5	20480	19.50	47.93	37.50	30.70	28.43

TABLE 2

1	2	3	4	5	6
R.P.M.	I.H.P. (LOADED)	Q CAPACITY IN GALLONS PER MINUTE	H TOTAL HEAD IN FEET OF WATER =	W.H.P. WATER HORSE POWER = $\frac{H.G.}{3300}$	PUMP EFFICIENCY = $\frac{W.H.P.}{I.H.P.}$
75	8	6000	2.25	4.264	.534
95	14	7600	4.15	8.147	.582
115	22.5	9200	5.9	14.025	.623
135	35.2	10850	7.93	22.2	.685
155	52	12450	10.33	33.28	.694
175	72	14060	13.00	47.25	.713
195	96.2	15660	16.37	66.25	.748
215	125.5	17260	20.2	90.1	.78
235	160.4	18860	24.2	118.0	.80
255	200.5	20480	28.6	151.3	.82

TABLE 3

1	2	3	4	5	6
R.P.M.	H TOTAL HEAD IN FEET OF WATER	Q CAPACITY IN GALLONS PER MINUTE	W.H.P. WATER HORSE POWER = $\frac{H.G.}{3300}$	I.H.P.	PUMP EFFICIENCY = $\frac{W.H.P.}{I.H.P.}$
50	2.3	2450	1.455	4.00	.39
100	6.5	4900	6.23	18.5	.4935
125	11.06	6650	14.00	44.5	.464
150	14.66	7750	21.05	71.0	.4445
185	19.59	9080	45.75	113.0	.44
210	25.1	10890	66.8	161.0	.451
225	28.7	11040	81.9	194.0	.439
240	32.3	11750	98.1	228.5	.4448

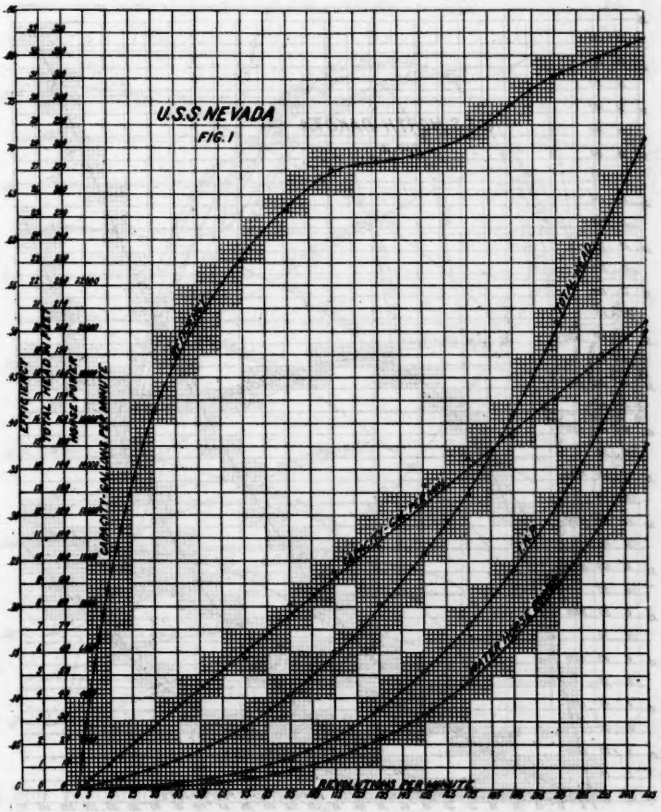
TABLE 4

1	2	3	4	5	6	7	8	9
R.P.M.	READING AT POINT No. 1	READING AT POINT No. 2	READING AT POINT No. 3	READING AT POINT No. 4	26.33 FT. MINUS READING AT POINT No. 1	DIFFERENCE BETWEEN READINGS OF POINTS No. 2 & No. 3	DIFFERENCE BETWEEN READINGS OF POINTS No. 3 & No. 4	READING OF POINT No. 4 MINUS 26.33 FT.
75	25.91	28.53	28.10	26.75	.42	.53	1.25	.42
95	25.57	29.50	28.46	27.68	.96	1.02	.80	1.35
115	24.70	30.73	29.10	28.06	1.63	1.63	1.04	1.73
135	24.20	32.30	29.80	28.60	2.13	2.50	1.20	2.27
155	23.90	34.23	30.70	28.80	2.43	3.53	1.90	2.47
175	23.90	36.78	31.60	30.00	2.43	5.18	1.60	3.67
195	23.10	39.48	32.85	30.45	3.23	6.63	2.40	4.12
215	21.90	42.33	34.60	30.65	4.43	7.73	3.95	4.32
235	20.70	45.03	36.00	30.66	5.63	9.03	5.34	4.33
255	19.50	47.93	37.50	30.70	6.83	10.43	6.80	4.37

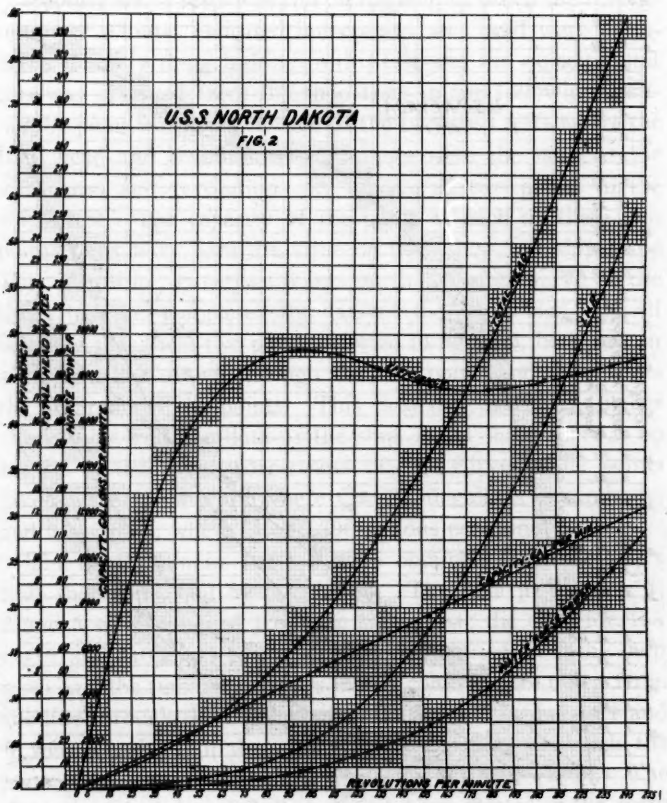
Another set of tests was run on the ship in which the revolutions per minute, indicated horsepower and head were measured; the only thing lacking in this test was the capacity, and this was obtained from the shop tests in the following manner: A point was taken in each of the two shop tests where the total head and revolutions per minute were the same as the total head and revolutions per minute of two points in the ship's test. The capacity of the pump at each of these two points must have been the same on both shop and ship tests. This, therefore, gave two points on the capacity curve of the ship's test and completed the data of the ship's test. It will be noted that these two points, plotted in figure 2, both fall on a straight line drawn through the zero point, and this checks the accuracy of the points. This capacity curve was further checked by trial-trip data of the ship, and I think there is no reason to doubt the correctness of this curve. The points taken were 158 r.p.m., where Q=7,750 gallons per minute, and 210 r.p.m., where Q=10,300 gallons per minute.

The data given in Table 3 were obtained from the ship's test, shop tests, and by calculation. The data in columns 1, 2 and 5 were obtained from the ship's test; the data for 158 r.p.m. and 210 r.p.m. in column 3 were obtained from the shop tests, and the remaining points in this column were taken from the line through these points given in figure 2; columns 4 and 6 were obtained in the same manner as that in which the same data were obtained in Table 2 for the *Nevada*. The data given in columns 2, 3, 4, 5 and 6 are plotted in the form of curves in figure 2.

It will be noted that the characteristics of all the curves in figure 2 are the same as similar curves in figure 1. There is, however, a remarkable difference between the two pumps as regards maximum efficiency. In the accompanying drawings are given the layout of pumps, plans of the pumps, and plans of the condensers. The *Nevada's* pump had a 11 inches×22 inches×10 inches engine; the *North Dakota,* 11 inches×22 inches×12 inches. From this it will be seen that there is



no very radical difference in the designs and arrangements for the two ships. When the *North Dakota* was placed in commission it was found that there was a sharp corner in the volute casting on the discharge side of the pump; this caused considerable vibration in the pump and when it was removed the vibration ceased; this small change, however, could not possibly account for the enormous difference in the efficiencies of the two pumps. The difference is due to the design of the pumps and the conditions under which they are operated. The



relation of capacity and head have so much bearing on the efficiency that it is possible for an excellent pump to give very poor results when operated under improper conditions of head and capacity. This point is illustrated in figures 4 and 5. Figure 4 gives curves plotted from actual test on a small capacity, low-head pump. Two conditions have been assumed; the first is with the pump delivering 2,000 gallons per minute at 1,350 revolutions per minute, and the second is with the pump delivering 800 gallons per minute when running at

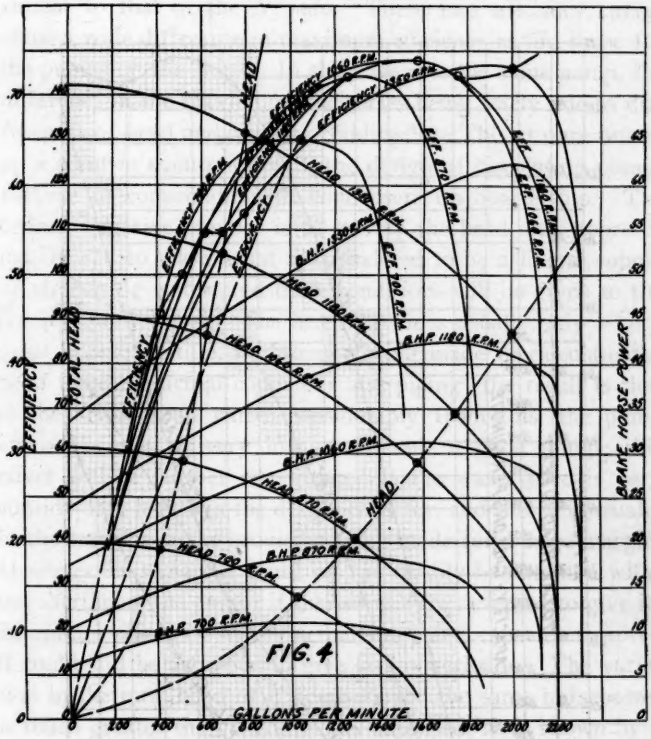


FIG. 4

1,350 revolutions per minute; it has been assumed that the capacity will vary directly as the revolutions in both cases—in other words, condenser conditions are assumed. With these conditions cross curves of head and efficiency have been plotted for both cases, and these curves are marked "head" and "efficiency" in figure 4. To make these curves comparable with those given in figures 1 and 2 they have been replotted with revolutions per minute as abscissae and are shown in figure 5. All curves shown agree with corresponding curves in figures 1 and 2, one of the efficiency curves being similar to that of the *North Dakota* and the other being

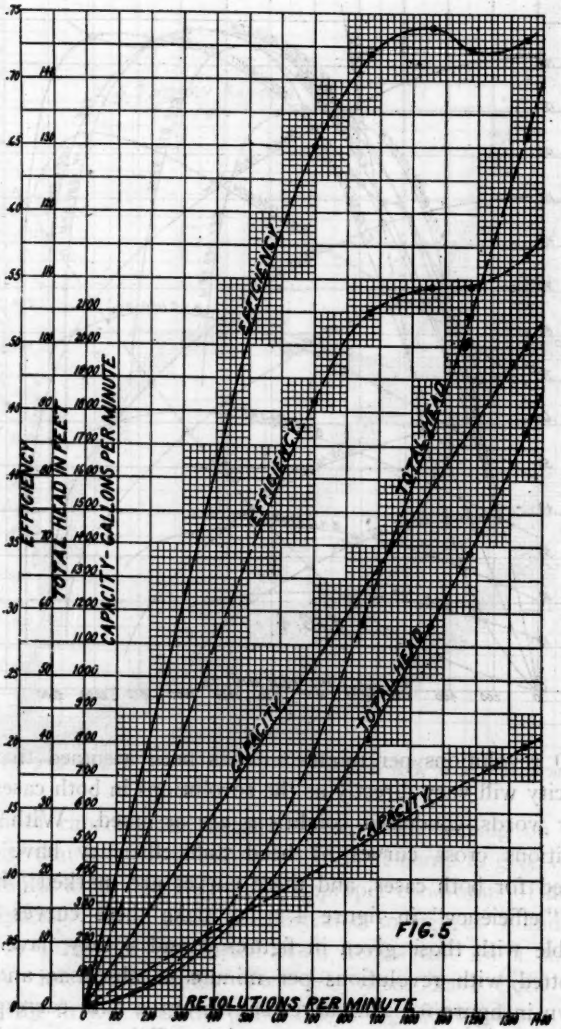

EFFICIENCY
TOTAL HEAD IN FEET
CAPACITY - GALLONS PER MINUTE
REVOLUTIONS PER MINUTE
EFFICIENCY
EFFICIENCY
TOTAL HEAD
TOTAL HEAD
CAPACITY
CAPACITY

FIG. 5

similar to that of the *Nevada*. These two efficiency curves show a wide difference in maximum efficiency as did those for the pumps on the ships. In this case it is the same pump, the difference in the maximum efficiencies being solely due to differences of head and capacity conditions. This at once brings up a point in connection with the design of circulating pumps for use in connection with condensers on board ship. The ordinary marine practice is to specify the head only approximately and to allow what is considered to be a liberal supply of circulating water, and these conditions will be given to the pump manufacturer. The head chosen is usually only a very great approximation, no attempt being made to calculate the head from the actual condenser and piping; the result is that if the actual head varies considerably from this, the pump efficiency may be very low, even with a good pump; this effect will be further exaggerated if the water used is very considerably less than the design calls for, and this will usually be the case due to the provision being made for a large margin. However, this margin could just as readily be obtained without sacrificing the pump; if the pump were designed to give its maximum efficiency at the actual head and capacity required it could still be arranged to give large overloads. The variation in the total head of condensers for the same horsepower is much greater than is commonly supposed, as is shown by a comparison of the total head curves of the *North Dakota* and *Nevada*. The total head at the same capacity is about three times as great for the *North Dakota* as it is for the *Nevada*. This is an extreme case, as the *North Dakota's* piping is entirely too small, but doubtless there are many other cases just as bad.

An attempt has been made to analyze to the various parts of the total head; the result is shown in Table 4. Columns 1 to 5, inclusive, give the data taken on the *Nevada's* test. The water line was 26.33 feet above point No. 1, so the suction head on the pump will be given by the differences between the readings at point No. 1 and 26.33 feet; these differences

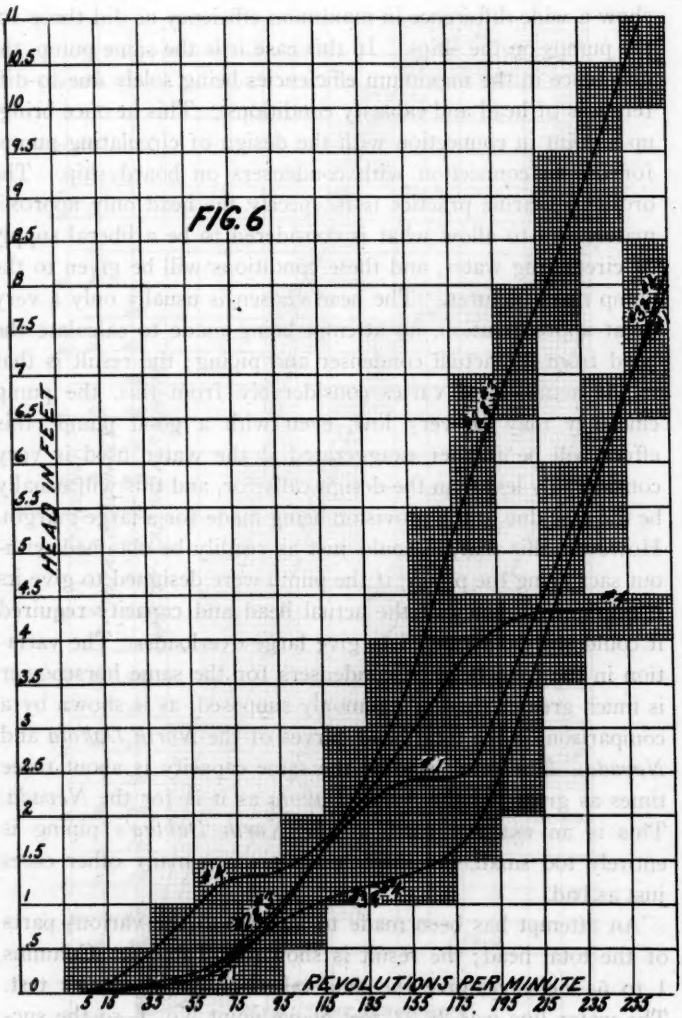
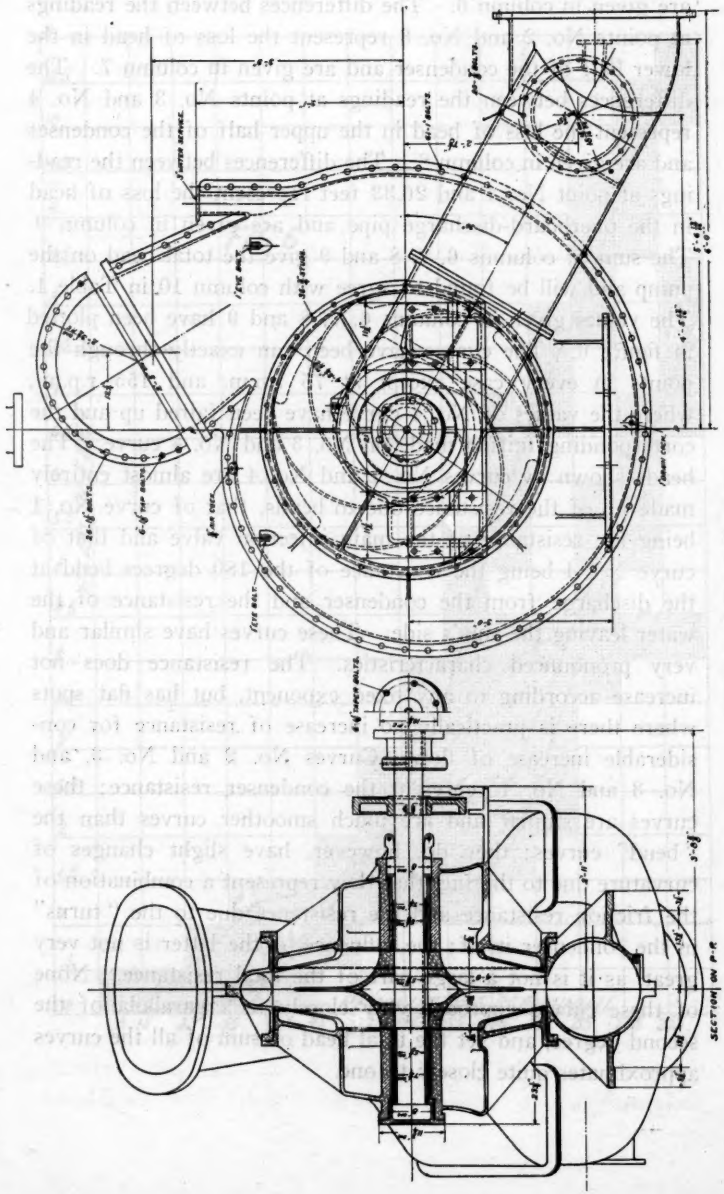
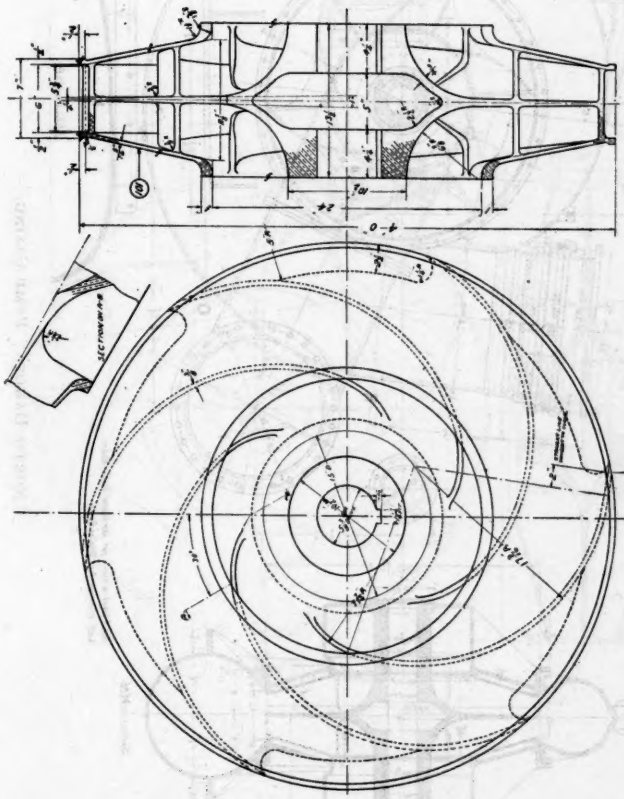


FIG. 6
HEAD IN FEET
REVOLUTIONS PER MINUTE
11
10.5
10
9.5
9
8.5
8
7.5
7
6.5
6
5.5
5
4.5
4
3.5
3
2.5
2
1.5
1
.5
0
5
15
25
35
55
75
95
115
135
155
175
195
215
235
255

are given in column 6. The differences between the readings at points No. 2 and No. 3 represent the loss of head in the lower half of the condenser and are given in column 7. The differences between the readings at points No. 3 and No. 4 represent the loss of head in the upper half of the condenser and are given in column 8. The differences between the readings at point No. 4 and 26.33 feet represent the loss of head in the overboard-discharge pipe and are given in column 9. The sum of columns 6, 7, 8 and 9 give the total head on the pump and will be found to agree with column 10 in Table 1. The values given in columns 6, 7, 8 and 9 have been plotted in figure 6. The curves have been run exactly through the points in every case except at 75 r.p.m. and 155 r.p.m., where the values of No. 4 curve have been faired up and the corresponding fairing made on No. 3 and No. 4 curve. The heads shown by curves No. 1 and No. 4 are almost entirely made up of the resistance due to bends, that of curve No. 1 being the resistance of the main injection valve and that of curve No. 4 being the resistance of the 180 degrees bend at the discharge from the condenser and the resistance of the water leaving the ship's side. These curves have similar and very pronounced characteristics. The resistance does not increase according to any fixed exponent, but has flat spots where there is practically no increase of resistance for considerable increase of flow. Curves No. 2 and No. 3, and No. 3 and No. 4 represent the condenser resistance; these curves are similar and are much smoother curves than the "bend" curves; they do, however, have slight changes of curvature due to the fact that they represent a combination of the friction resistance and the resistance due to the "turns" in the condenser itself; the influence of the latter is not very great, as it is not a large part of the total resistance. None of these curves approach very closely to a parabola of the second degree, and yet the total head or sum of all the curves approximates quite closely to one.

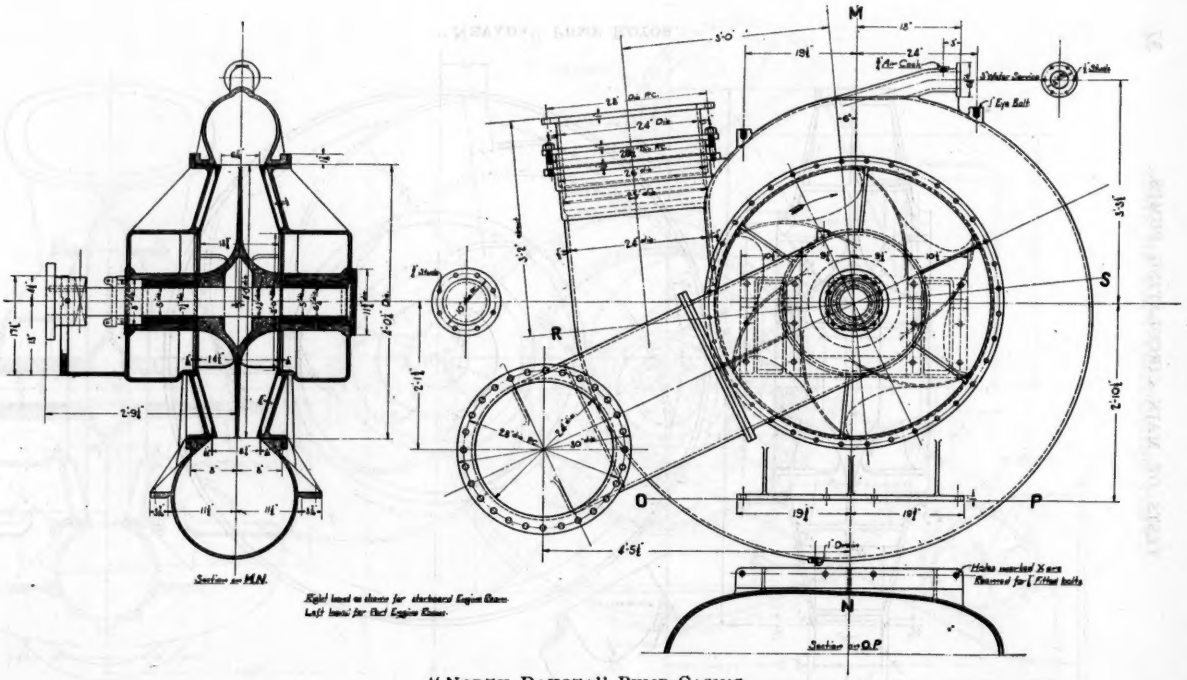


"NEVADA" PUMP CASING.

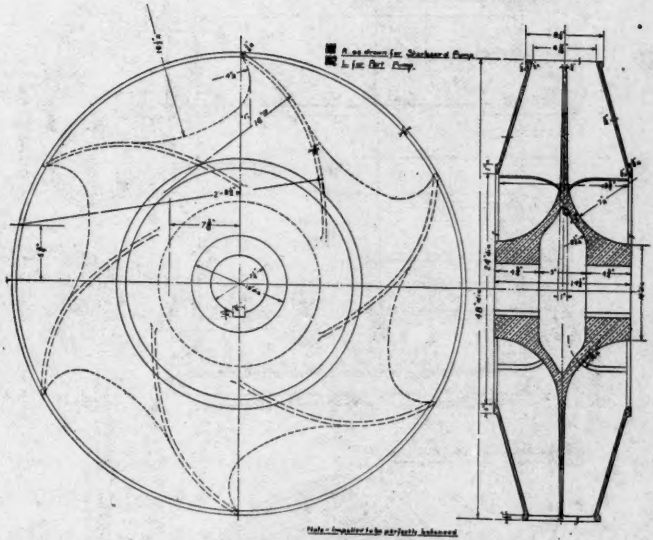


"NEVADA" PUMP CASING.

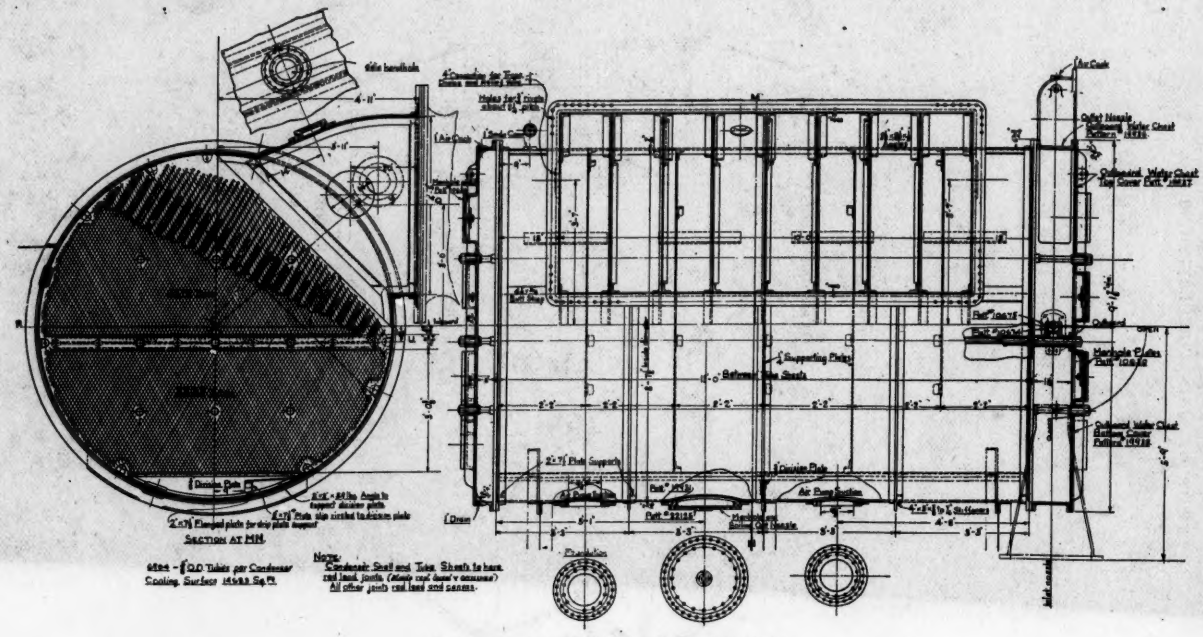
"NEVADA" PUMP ROTOR.



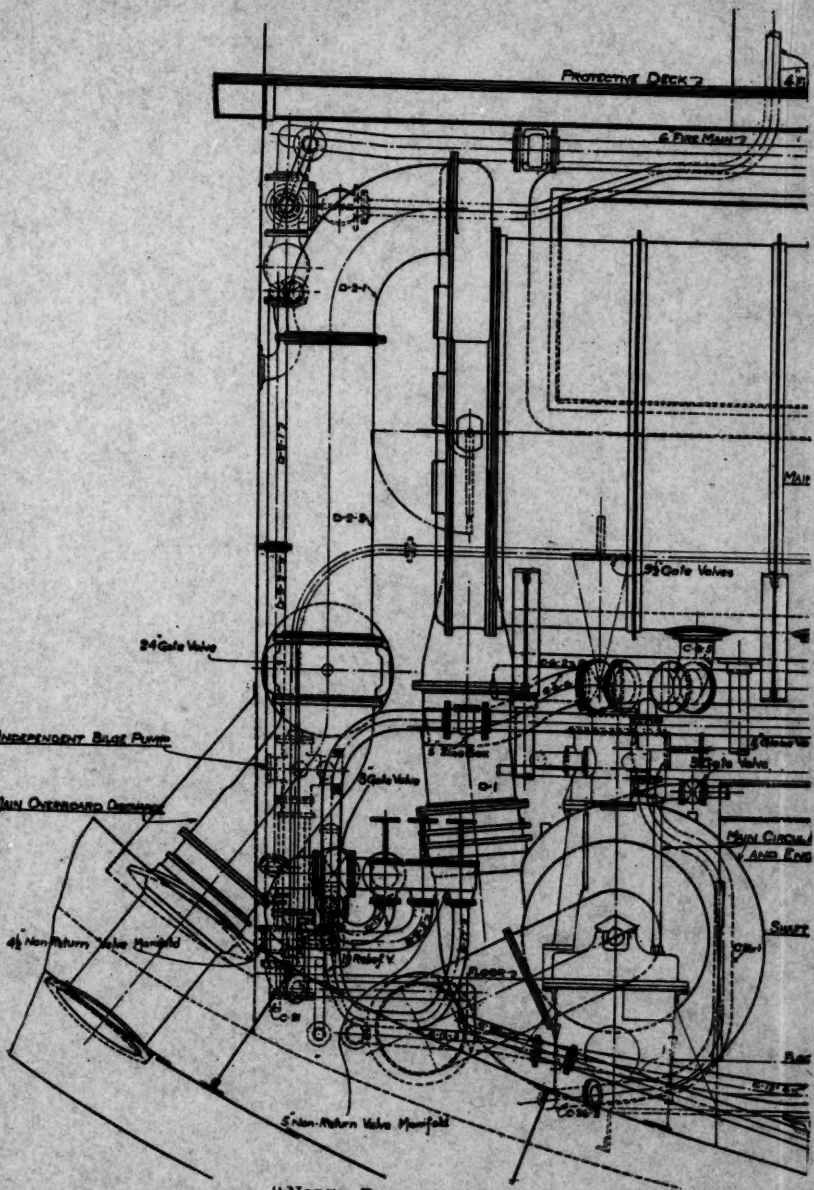
"NORTH DAKOTA" PUMP CASING.



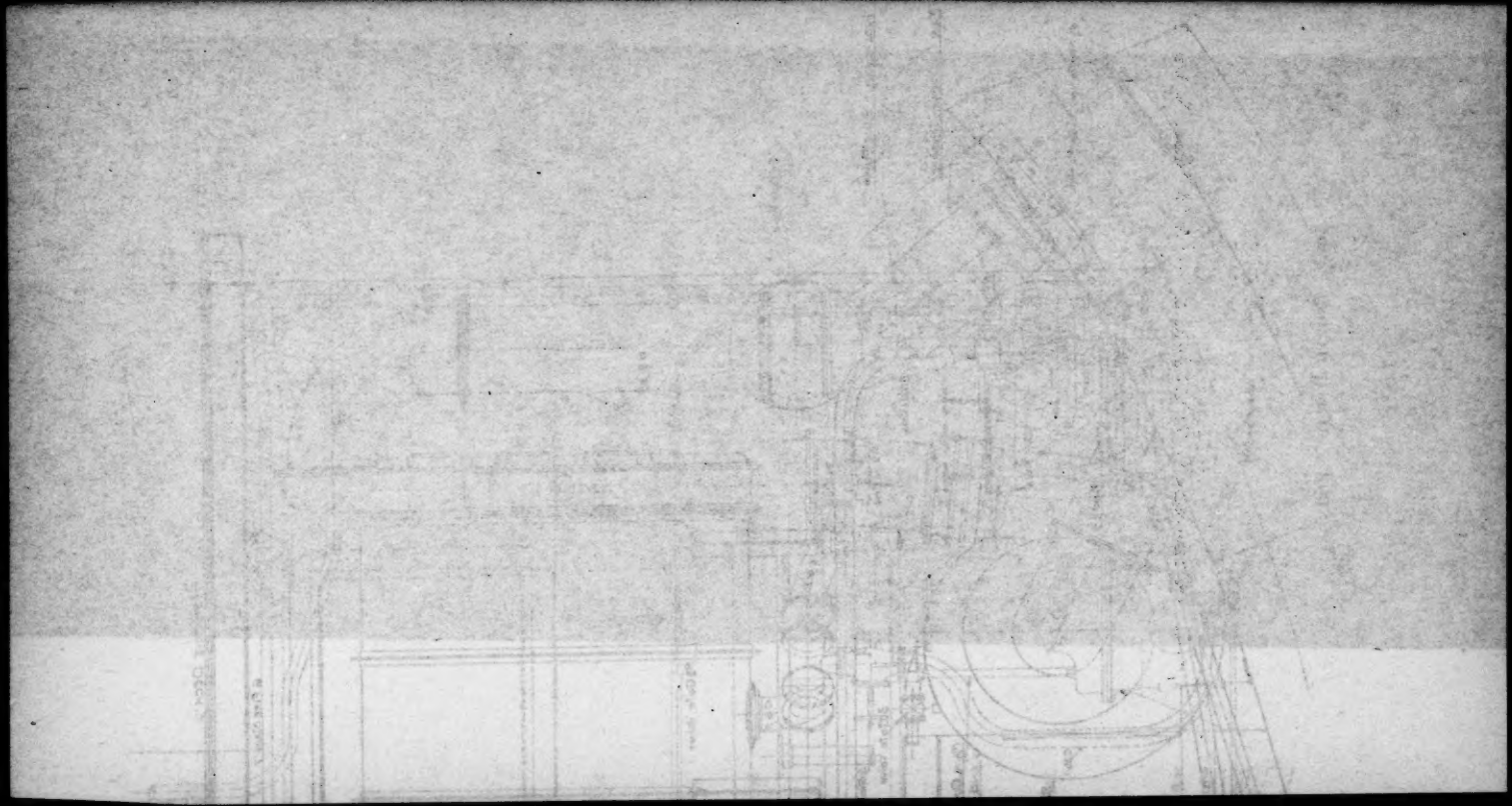
"NORTH DAKOTA" PUMP ROTOR.

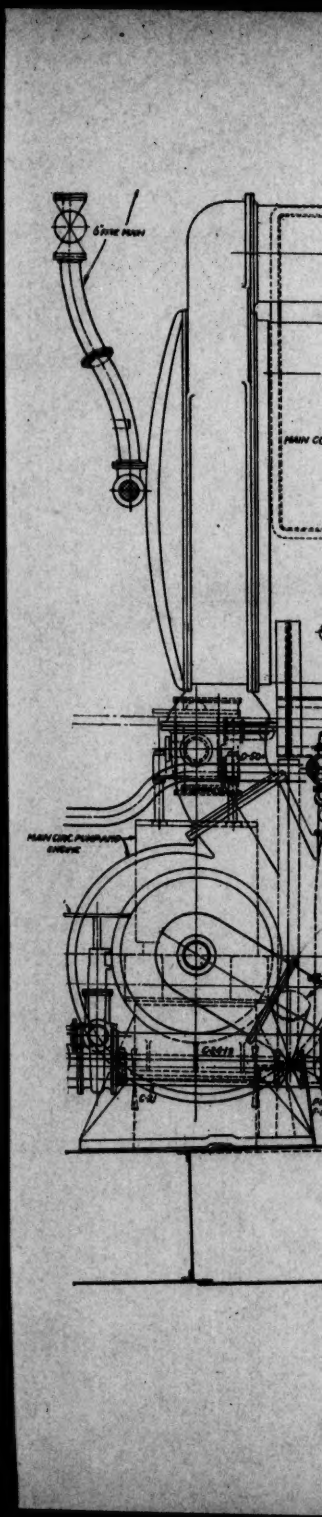


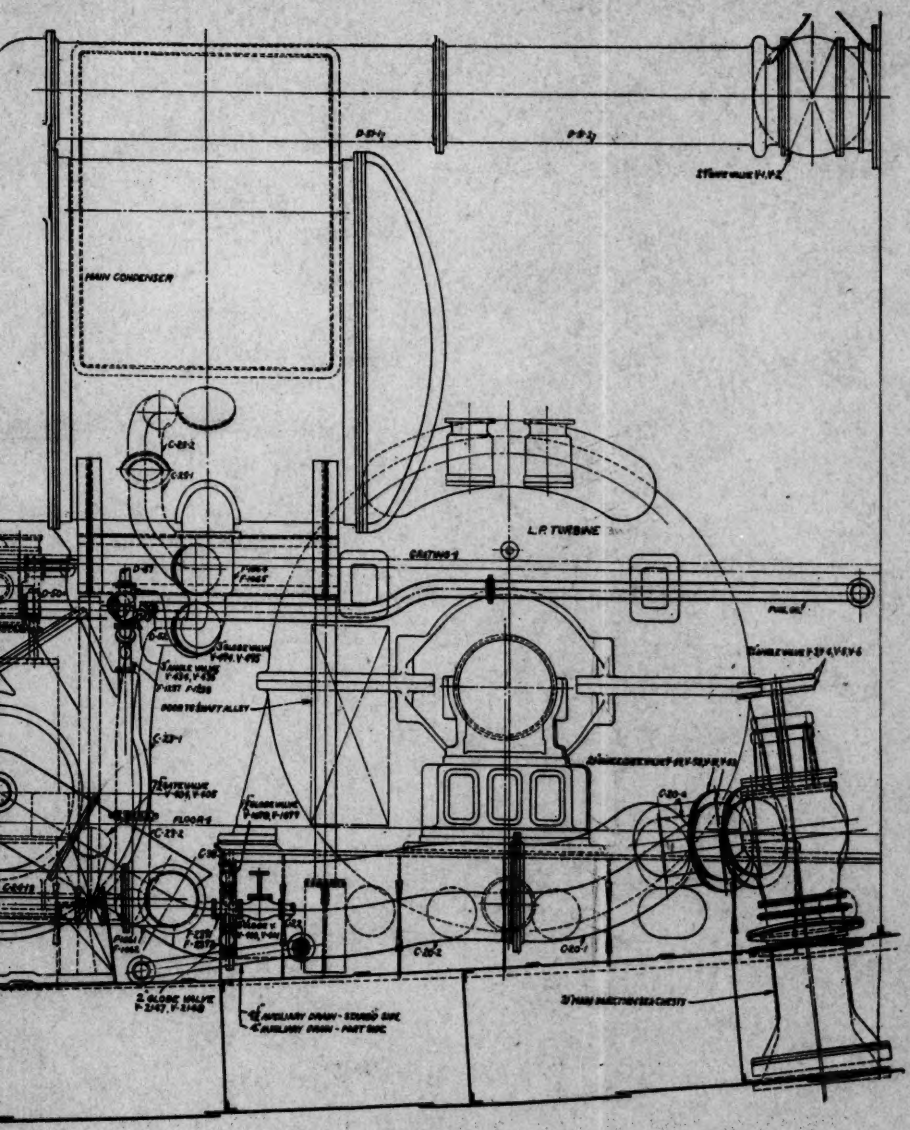
"NORTH DAKOTA" MAIN CONDENSER.



"NORTH DAKOTA" PIPING.

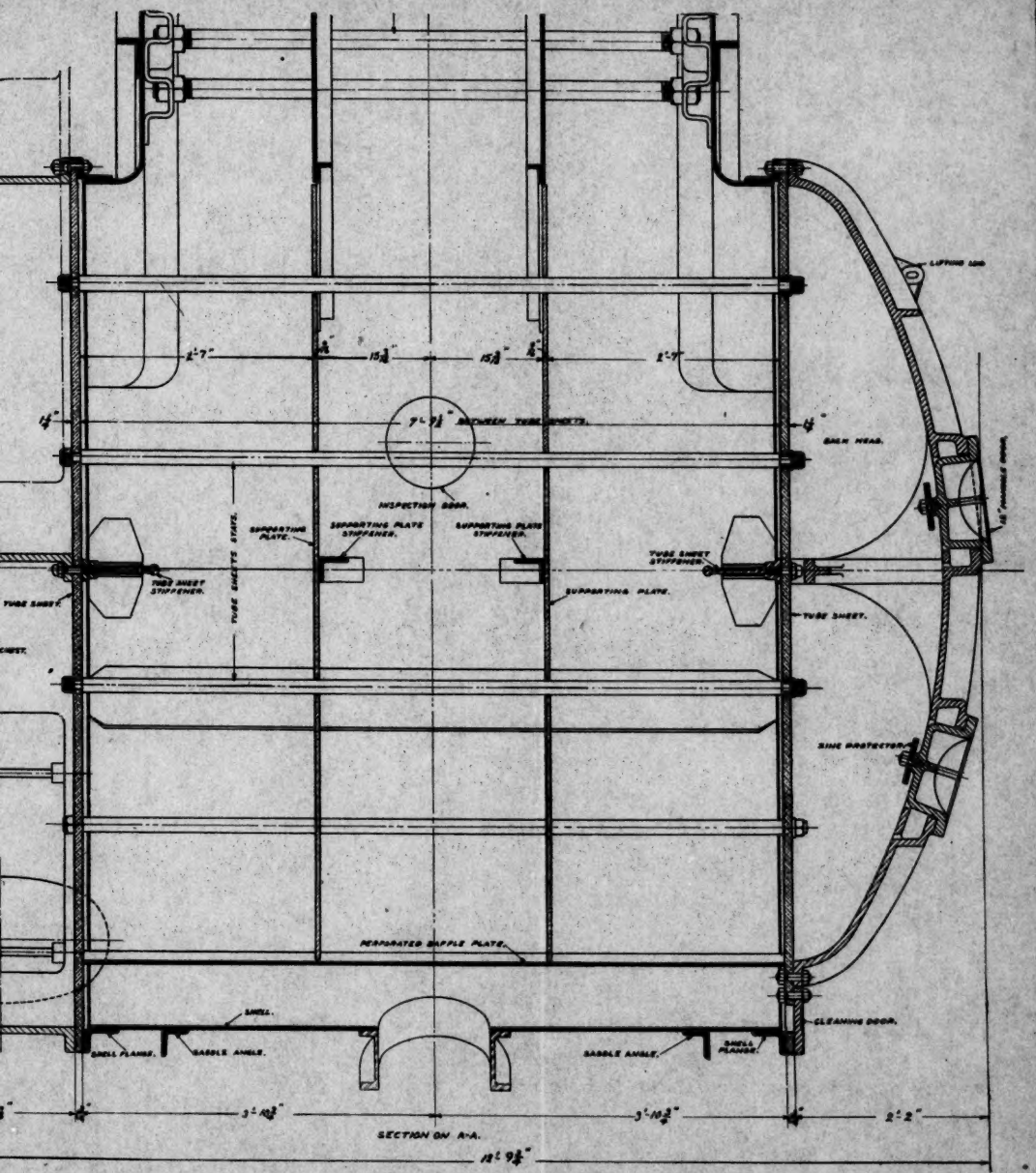




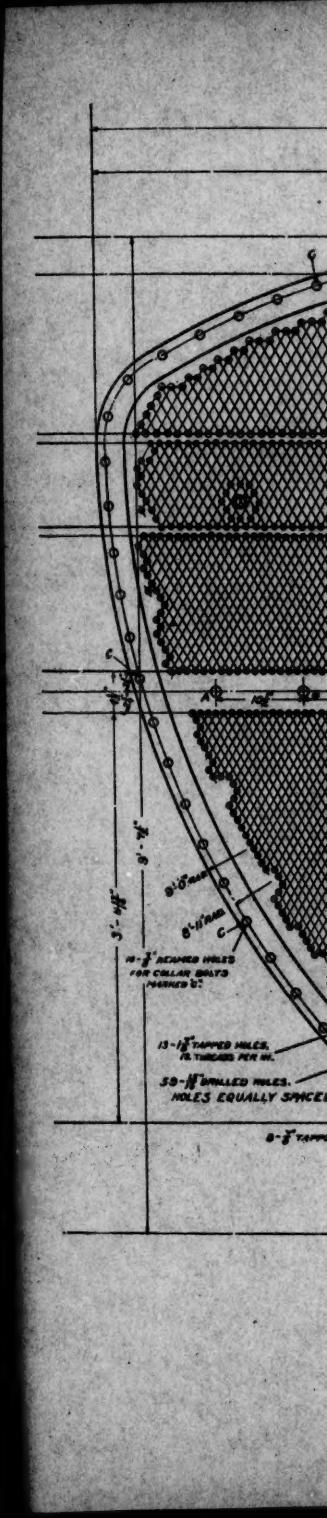


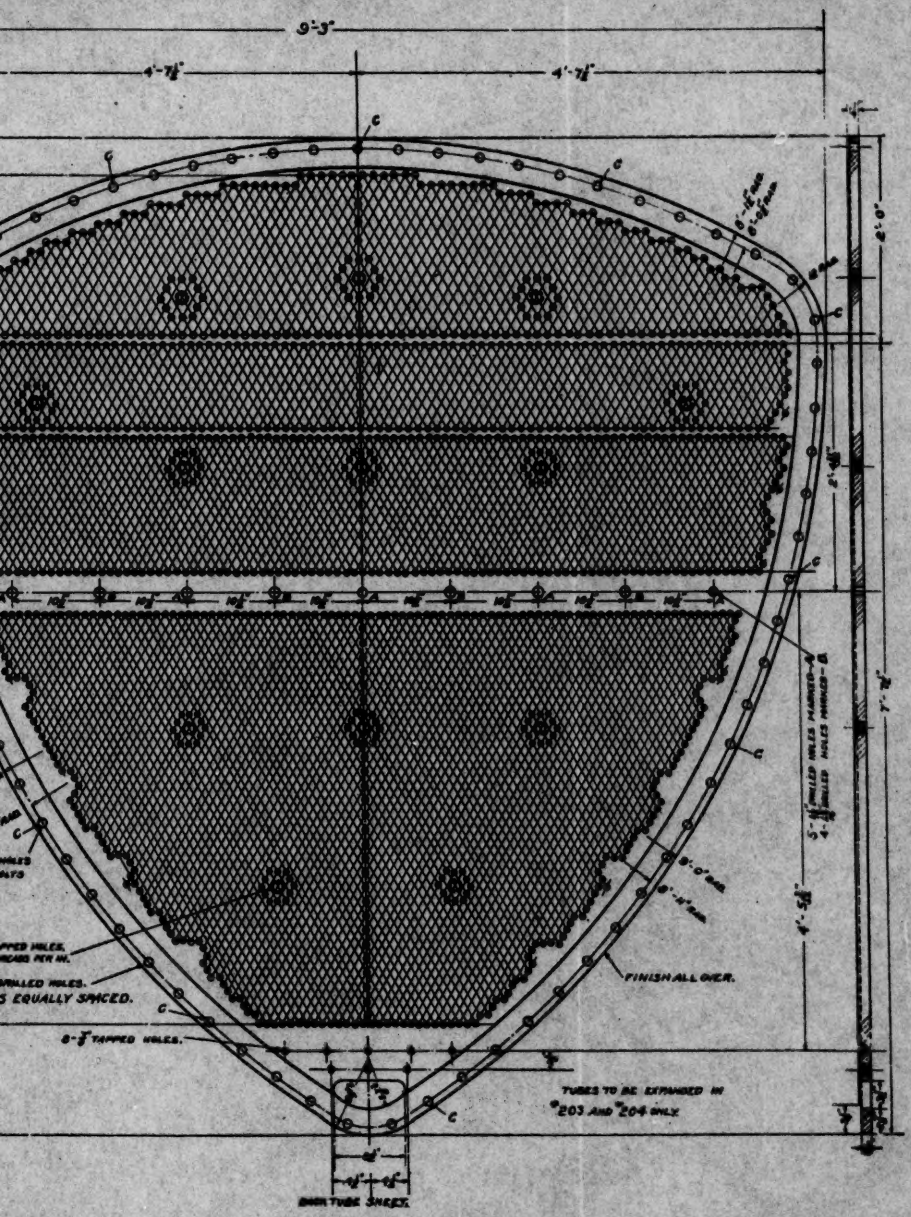
"Nevada" Piping.

LIFTING LUG.
ZINC PROTECTOR.
TUBE SHEET.
WATER CHEST.
FRONT HEAD.
12"
11 3/4"



"NEVADA" CONDENSER.


C
A
10"
C
8'-0" RAD.
8'-11" RAD.
C
18-⅞" REAMED HOLES
FOR COLLAR BOLTS
MARKED 'B'
13-1⅛" TAPPED HOLES,
12 THREADS PER IN.
59-⅞" DRILLED HOLES.
HOLES EQUALLY SPACE



"NEVADA" CONDENSER.

TEST OF REGENERATIVE EVAPORATING APPARATUS.

By M. C. Stuart, Associate.

A test was made recently at the U. S. Naval Engineering Experiment Station, Annapolis, Md., upon a patented system for the regenerative operation of evaporators, submitted by the Regenerative Evaporator Co., of Baltimore, Md., in which the efficiency of multiple effect operation was obtained with a single evaporator shell. This article will include a description of the system, description and results of the test, and calculations for the performance and installation of the system based on the test results.

DESCRIPTION OF APPARATUS.

The essential feature of the system consists of a compressor, called a "Jetflo" compressor, the function of which is to compress, by the use of live steam, part of the vapor produced by the evaporator, and to return to the coils of the evaporator the compressed vapor together with the steam used in compressing it. The action of the system may be described by reference to the photograph of the Jetflo compressor, Fig. 1, and to the drawing, Fig. 2, showing the application of the compressor to the evaporator upon which the test was conducted. Referring to Fig. 1, the compressor is operated by high-pressure steam entering at S. The side opening V is connected to the vapor line from the evaporator, and the bottom opening E is connected to the inlet of the coils in the evaporator. Steam, entering at S, passes through a strainer at A and expands through nozzles in the compressor body B. At the outlet of the nozzles the steam, by reason of its high velocity, entrains a certain amount of vapor, and the steam

and entrained vapor, continuing through a diffuser tube, also in the body B of the compressor, are compressed, entering the steam coil at E with a pressure higher than the original vapor pressure generated in the shell, but lower than the pressure of the steam supplied. The steam and entrained vapor are condensed in the coils and removed by the trap shown in Fig. 2. The vapor which is not compressed, herein referred to as the non-entrained vapor, is discharged from the vapor outlet and condensed in the distiller condenser as usual.

The by-pass D, Fig. 1, is fitted with a valve (3) and an orifice plate C for use when it is desired to cut out the compressor and operate the evaporator in the usual manner with single-effect efficiency. The orifice is of such size as to limit the amount of steam at the pressure available to that required for producing the maximum safe capacity of the evaporator. The gage G, Fig. 1, serves both to indicate the steam pressure and to show the steam consumption with the compressor or the by-pass operating. The steam flow through either the compressor or the by-pass orifice is nearly proportional to the absolute steam pressure, and the outer scale is laid off so that when multiplied by a suitable constant it represents the flow of steam through the compressor or the by-pass orifice. The inner scale is graduated to show the steam pressure in pounds per square inch gage.

OBJECT AND METHOD OF TEST.

The efficiency of the system is primarily a function of the amount of vapor compressed, because each pound of vapor compressed and returned to the coils practically replaces one pound of steam which would otherwise be required to produce the same amount of evaporation of fresh water. The capacity of the system is a function of the amount of compression produced by the compressor, because this determines directly the temperature difference which will exist between the coils and the shell. The principal objects of the test were, therefore, to determine the following quantities for various condi-

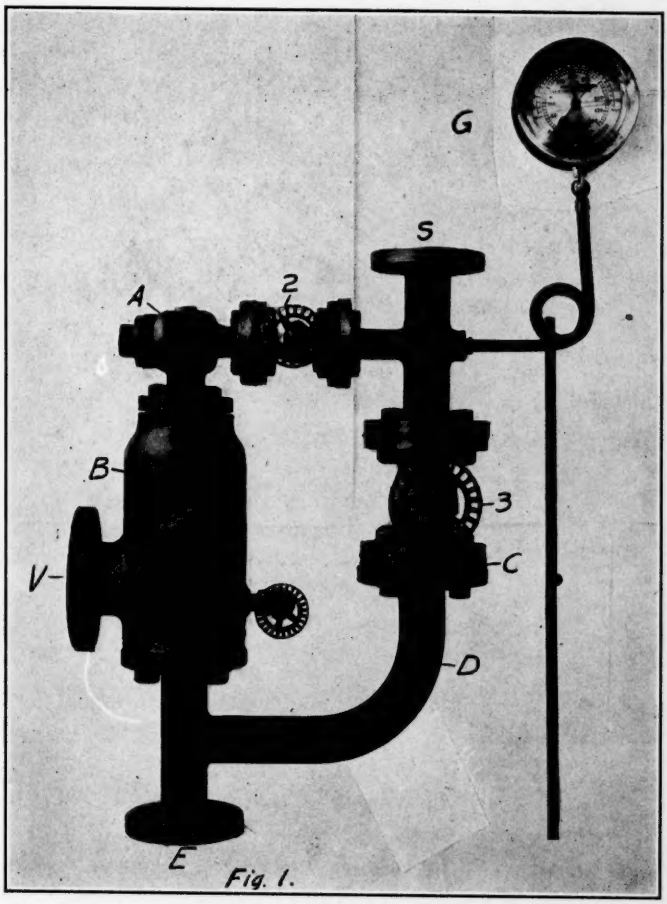


Fig. 1.

tions of operation: Steam used by the compressor, vapor entrained and compressed, non-entrained vapor produced, total vapor produced, amount of compression produced by the compressor, and temperature difference maintained between coils and shell.

The arrangement of the apparatus for test was practically

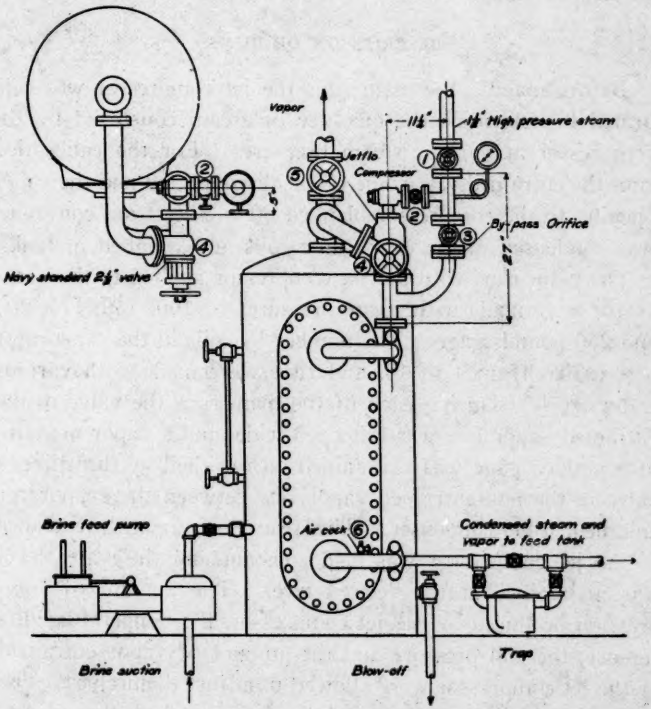


FIG. 2.

as shown in Fig. 2. The coil condensate, consisting of steam and entrained vapor, was cooled and weighed. The steam pressure was measured with a calibrated gage. The vapor pressure in the shell and the amount of compression produced by the Jetflo compressor were measured with mercury mano-

meters. Vertical connections to the mercury manometers were made with glass tubing in order that the exact height of water over the mercury could be observed and the proper correction applied to obtain the true pressures. A separator was placed in the steam line to remove excess moisture, and a throttling calorimeter was connected beyond the separator to determine the quality of steam supplied to the compressor.

DESCRIPTION OF RUNS.

Before making the main runs the jet compressor was calibrated to determine the quantity of steam consumed by the compressor at various steam pressures. For the calibration runs the entrained vapor line C was disconnected and the vapor opening to the compressor blanked off. The steam consumed was condensed in the evaporator coils and weighed in tanks.

The main runs with the jet compressor applied to the evaporator were made with steam pressures of 100, 150, 175, 200 and 250 pounds gage. The number of coils in the evaporator was varied from 4 to 12, and runs were made with various water levels. During each of the main runs the valve in the entrained vapor line was left open wide, and a vapor pressure of 5 pounds gage was maintained on the shell by throttling a valve in the non-entrained vapor line between the evaporator and the distiller condenser. The steam pressure at the entrance to the jet compressor was held constant and the water level was maintained at the desired level. The amount of compression produced by the jet compressor, the temperature difference, the coil pressure and the amount of vapor entrained by the jet compressor were allowed to adjust themselves. The following discussion indicates the manner in which this adjustment takes place.

The heat supplied to the coils and available for transmission through the heating surface is a function of the amount of steam used by the compressor plus the amount of vapor entrained by the steam. The amount of heat actually transmitted by the heating surface and taken up by the water in the

Test No	ITEM	UNIT	SOURCE	NOZZLE TIP No 1															
				8	9	11	12	14	16	27	28	29	30	31	32	33	34	35	
1	Run Number		Test Data							5-10	5-10	5-10	5-11	5-12	5-14	5-15	5-16	5-18	
2	Date Of Run - 1917		Test Data	4-25	4-25	4-26	4-26	5-1	5-3										
3	Duration Of Run	Minutes	Test Data	60	60	60	60	60	60	60	60	60	60	60	60	60	90	60	
4	Number Of Coils In Shell		Test Data	8	8	8	8	6	12	12	12	12	12	10	8	6	8	8	
5	Room Temperature	Deg. Fahr.	Test Data	72.0	74.0	73.0	70.0	69.9	73.9	72.0	72.9	73.9	73.0	77.0	85.0	84.0	78.9	78.0	
6	Water Level Above ℄ Bottom Manifold	Inches	Test Data	20.5	20.5	20.5	20.5	20.5	20.5	19.9	15.0	10.0	5.0	22.5	18.7	26.5	20.4	20.9	
7	Evaporating Surface Installed	Sq. Ft.	See Appendix II	36.206	36.206	36.206	36.206	27.0.710	36.478	36.478	36.478	36.478	36.478	32.542	24.205	20.070	36.206	36.205	
8	Steam Pressure, Gage, At Compressor	Lb. Per Sq. In.	Test Data	125.0	125.0	125.0	125.0	125.0	124.7	174.1	174.1	174.1	174.1	174.1	174.1	174.0	174.1	164.5	
9	Barometer	Inches Hg.	Test Data	29.00	29.97	29.98	29.97	29.71	29.99	29.91	29.90	29.87	29.78	29.74	29.87	29.88	30.00	29.97	
10	Pressure In Calorimeter, Gage	Lb. Per Sq. In.	Test Data	14.4	14.2	14.4	14.8	14.7	14.8	22.4	22.2	22.1	22.3	22.5	22.1	22.0	18.2	16.4	
11	Temperature In Calorimeter	Deg. Fahr.	Test Data	305.9	305.7	305.0	305.4	303.1	307.0	303.0	303.6	305.6	302.6	302.0	301.5	302.6	298.5	300.5	
12	Total Heat In Steam	B.t.u. Per Lb.	See Appendix II	1191.2	1191.5	1191.0	1191.2	1189.5	1191.9	1187.9	1188.1	1189.0	1187.9	1187.2	1187.4	1188.0	1189.8	1188.0	
13	Temperature Of Steam At Compressor	Deg. Fahr.	Test Data	366.9	366.7	366.1	366.1	364.9	367.4	376.2	375.6	373.8	373.6	377.0	378.9	378.5	373.5	374.9	
14	Superheat In Steam At Compressor	Deg. Fahr.	See Appendix II	4.5	3.9	3.3	3.9	1.6	4.7										
15	Quality Of Steam At Compressor	Per Cent Dry	See Appendix II							98.9	98.9	99.0	98.9	98.8	98.9	98.9	98.8	98.0	
16	Quality Of Steam At 250 Lb. Gage	Per Cent Dry	See Appendix II	98.7	98.7	98.6	98.7	98.4	98.7	98.9	98.5	98.4	98.9	98.2	98.2	98.9	98.1	98.9	
17	Weight Of Condensate From Coils	Lb. Per Hr.	Test Data	928.0	916.9	878.5	880.0	648.0	978.5	991.0	989.5	967.9	969.0	999.5	995.5	907.0	949.8	908.0	
18	Weight Of Non-Entrained Vapor	Lb. Per Hr.	Test Data	84.6	84.9	86.9	86.2	98.0	80.1	81.4	77.3	85.1	87.9	78.8	90.8	88.2	92.0	86.9	
19	Weight Of Steam Used By Compressor	Lb. Per Hr.	See Appendix II	1012.9	1001.7	1005.4	1001.9	1092.4	1092.2	1072.4	1067.0	1053.0	1056.9	1078.7	1086.3	1096.0	1041.8	1094.9	
20	Weight Of Entrained Vapor	Lb. Per Hr.	(17) - (19)	172.7	176.2	173.2	178.7	91.6	217.9	248.0	248.5	230.9	225.1	246.6	208.5	190.1	198.5	171.9	
21	Weight Of Total Vapor	Lb. Per Hr.	(20) + (18)	258.7	260.5	259.5	258.9	191.6	297.4	329.4	325.8	316.0	313.0	325.4	287.4	278.6	290.5	257.8	
22	Lb. Of Entrained Vapor Per Lb. Of Steam Used		(20) ÷ (19)	1.1829	1.1729	0.8996	0.8929	0.8076	1.4994	1.4667	1.4487	1.5009	0.8912	1.0787	1.4048	1.0899	1.9988	1.2478	
23	Lb. Of Total Vapor Per Lb. Of Steam Used	Lb. Per Hr.	(21) ÷ (19)	1.7392	1.7391	1.8009	1.4369	1.2572	2.0197	2.2204	2.1759	2.0685	2.1195	2.2151	1.9281	1.8719	1.9646	1.8899	
24	Total Vapor And Condensed Steam	Lb. Per Hr.	(17) + (18)	403.6	410.8	378.8	366.2	344.0	458.6	473.4	466.8	450.0	396.9	472.9	454.4	391.5	438.8	395.9	
25	Total Vapor And Condensed Steam Per Lb. Of Steam Used	Lb. Per Hr.	(24) ÷ (19)	2.7182	2.7392	2.3609	2.4369	2.2572	3.0197	3.2204	3.1759	3.0693	3.1139	3.2151	2.9851	2.6719	2.9646	2.8149	
26	Temperature Of Feed	Deg. Fahr.	Test Data	68.8	72.9	72.2	68.2	69.8	72.8	72.0	72.9	72.2	73.4	81.0	74.1	72.0	77.7	75.0	
27	Temperature Of Vapor	Deg. Fahr.	Test Data	227.5	227.4	227.3	226.8	226.7	227.2	226.2	226.4	226.7	226.6	226.6	226.7	226.5	226.9	227.0	
28	Vapor Pressure, Gage	Lb. Per Sq. In.	See Appendix II	5.05	5.15	4.96	4.84	4.95	5.04	484	484	502	487	5.00	4.94	4.97	4.94	5.02	
29	Press. Diff. Between Coil & Shell	Lb. Per Sq. In.	See Appendix II	6.91	7.05	7.20	7.64	8.26	5.49	6.00	6.05	6.76	5.48	6.01	7.19	8.05	7.96	7.19	
30	Coil Pressure, Gage	Lb. Per Sq. In.	(28) + (29)	11.96	12.20	12.66	12.48	13.21	10.47	1084	1082	1178	1035	11.01	12.19	13.02	12.90	12.15	
31	Temperature At Coil Entrance	Deg. Fahr.	Test Data	244.7	245.9	245.5	245.3	250.9	245.0	244.0	244.1	246.4	250	244.0	248.4	254.6	249.2	250.4	
32	Superheat At Coil Entrance	Deg. Fahr.	See Appendix II	1.0	1.5	4.3	3.9	4.7	2.1	2.8	2.7	5.2	4.5	2.6	4.4	5.8	4.7	6.9	
33	Compression Produced By Compressor, Gage	Lb. Per Sq. In.	See Appendix II	7.04	7.86	7.78	7.67	8.43	5.72	6.07	6.07	6.87	5.58	6.12	7.26	8.09	7.81	7.08	
34	Pressure Losses In Vapor Lines	Lb. Per Sq. In.	(33) - (29)	0.13	0.09	0.08	0.03	0.05	0.09	0.07	0.09	0.11	0.07	0.11	0.04	0.04	0.05	0.00	
35	Temp. Diff. Between Coil & Shell	Deg. Fahr.	See Appendix II	16.9	16.8	16.1	16.8	19.8	13.8	14.5	14.7	16.1	18.9	14.8	17.1	18.8	17.4	15.8	
36	Heat Given To Vapor	B.t.u. Per Hr.	See Appendix II	389964	390669	380009	296822	211971	440196	342846	341221	335689	296898	309886	282088	272946	262084	236098	
37	Temperature Of Coil Drain	Deg. Fahr.	Test Data	241.9	241.6	243.2	242.7	244.0	239.2	238.1	238.9	240.7	244.9	239.8	241.5	242.9	241.5	241.5	
38	Sub-Cooling Of Drain	Deg. Fahr.	See Appendix II	2.8	2.6	2.0	2.1	2.3	2.7	2.7	2.5	2.9	2.2	2.6	2.7	2.9	3.0	2.6	
39	Heat Of Entrained Vapor Above Drain Temp.	B.t.u. Per Hr.	See Appendix II	166140	166790	152004	122819	87470	178821	232451	230011	206382	185140	230949	198390	184191	186195	162256	
40	Heat Of Steam Above Drain Temp.	B.t.u. Per Hr.	See Appendix II	147304	147998	147204	147308	146910	149867	144232	144192	144050	145257	143977	143787	143477	143604	134855	
41	Heat Equivalent Of Work Of Compression Of Entrained Vapor	B.t.u. Per Hr.	See Appendix II	4964.8	4788	3984.7	3112.8	2338.6	4846.0	5248.5	5088	3095.6	3365	5279	5006	4986	4984	4047	
42	Gross Heat Of Entrained Vapor & Steam	B.t.u. Per Hr.	(39)+(40)	315644	314788	307107	270122	236380	358708	376702	374200	352804	330505	377205	342727	324708	329588	296799	
43	Net Heat Of Entrained Vapor & Steam	B.t.u. Per Hr.	(42)-(41)	311579	310078	303123	267009	233982	354162	371484	369004	347750	326861	372986	338081	320589	324884	292484	
44	Radiation & Unaccountable Losses	B.t.u. Per Hr.	(43)-(36)	21446	19438	15923	16487	23481	20148	8638	8189	10858	20888	12848	10909	11098	4238	3617	
45	B.t.u. Heat Transferred Per Sq. Ft. Of Evaporating Surface Per Deg. T.D.	B.t.u. Per Hr.	See Appendix II	7.39.0	7.12.8	8.60.6	6.62.9	6.91.8	6.90.2	6.63.6	6.61.4	6.01.4	6.82.9	7.84.7	7.43.4	7.07.6	7.12.5	6.64.5	
46	Radiation Loss Per Cent Of Heat Supplied Coils	Per Cent	(44)÷(43) X 100	6.88	6.26	5.25	6.18	9.91	5.68	2.33	2.16	3.12	6.30	3.45	4.18	3.87	1.31	1.23	
47	Salinity During Run	32nds	Test Data	0.40	0.56	1.70	1.73	1.72	1.40	1.38	1.60	2.06	2.10	1.24	1.38	1.71	2.16	2.27	
48	Lb. Of Non-Entrained Vapor Per Lb. Of Steam Used		(18)÷(19)	0.0835	0.0844	0.0827	0.0735	0.0496	0.0763	0.0769	0.0759	0.0869	0.0894	0.0964	0.0809	0.0850	0.0929	0.0826	
49	K5 = B.t.u. Per Hr. Per Deg. T.D.	B.t.u. Per Hr.	See Appendix II	19109	18680	14092	14354	11992	18783	23883	23402	21400	17808	25709	19894	19445	18671	17408	

FIG. 3.

NOZZLE TIP Nº 2

33	34	35	36	37	38	39	40	41	42	43	44	45	46	47	48	54	55	56	57	59	60	61	62
5-18	5-18	5-18	5-18	5-18	5-19	5-21	5-21	5-21	5-21	5-22	5-22	5-23	5-23	5-23	5-23	5-26	5-26	5-26	5-28	5-28	5-29	5-29	5-29
60	90	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	90	60	60	60
6	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	12	12	12	12	12	12	12	12
84.0	78.9	78.0	84.0	81.5	80.5	78.0	78.5	78.5	78.0	80.5	82.5	78.0	78.0	78.5	76.0	61.0	61.0	81.0	78.0	84.0	77.5	80.0	80.0
22.5	22.4	22.9	21.0	22.8	21.0	21.0	21.9	22.9	22.9	22.9	8.0	8.0	8.0	8.0	8.0	22.5	21.0	21.0	21.0	10.0	8.0	8.0	21.0
2.0070	26.205	26.205	26.205	26.205	26.206	26.205	26.205	26.205	26.205	26.205	26.205	26.205	26.205	26.205	26.205	26.205	26.205	26.205	26.478	26.478	26.478	26.478	26.478
17.40	17.41	16.15	18.67	19.94	14.68	17.41	17.41	17.41	17.40	28.8	28.6	14.88	19.94	23.00	14.89	17.85	14.29	19.86	20.17	29.14	20.12	29.6	20.61
23.88	30.00	23.97	23.94	30.00	23.99	23.84	23.95	23.95	23.75	23.71	23.98	23.60	23.98	23.86	23.71	23.99	23.97	23.94	23.64	23.80	23.71	23.74	23.77
22.0	18.2	16.4	22.5	22.2	18.1	21.4	21.9	21.9	21.9	18.1	12.1	18.2	26.7	21.6	18.2	22.2	21.8	18.4	22.3	24.7	26.4	21.6	21.4
30.15	29.15	30.05	30.06	29.48	29.81	30.76	30.14	30.46	30.43	29.12	29.48	30.11	30.61	30.07	30.09	30.15	30.58	29.69	30.62	30.82	30.0	30.06	30.86
118.60	118.68	118.60	118.90	118.40	118.74	118.66	119.00	119.01	118.98	118.71	118.66	118.81	118.61	118.71	118.77	119.01	119.07	118.73	118.91	118.78	119.06	118.79	119.00
57.25	57.25	56.99	58.43	56.88	56.43	58.35	57.59	57.62	40.42	54.14	54.07	56.44	58.69	40.50	56.42	57.70	56.86	54.81	58.17	40.86	58.19	40.87	58.17
										4.5	5.5								5.5				
98.9	98.8	99.0	98.9	98.2	99.2	98.9	99.1	99.2	98.0			99.2	99.7	98.1	99.2	99.1	98.5		98.8	98.3	98.9	98.9	98.9
98.5	98.1	98.5	98.4	97.8	98.2	98.0	98.6	98.8	98.0	98.1	98.1	98.5	98.9	98.1	98.2	98.8	98.6	98.5	98.4	98.2	98.3	98.5	98.5
307.0	349.8	309.0	307.0	498.5	303.5	98.0	496.0	223.0	468.0	155.5	24.5	7.7.0	174.5	256.0	168.0	419.5	97.2.5	218.5	458.3	408.8	57.2.5	405.5	497.9
68.5	91.0	68.0	68.8	101.0	75.0	98.2	98.7	100.1	139.6	67.0	12.9	21.9	154.1	135.8	92.4	78.7	72.9	95.0	102.4	140.4	114.8	141.6	108.0
144.9	147.0	192.5	136.4	166.0	178.0	146.9	146.9	148.8	209.2	86.5	68.3	172.8	166.0	208.9	127.9	148.0	118.7	88.9	167.8	204.9	167.1	208.1	163.0
160.1	196.5	178.5	210.6	259.5	174.5	208.1	149.1	84.1	143.8	27.0	0.0	0.0	8.5	98.1	55.1	268.9	249.8	190.2	208.0	204.9	198.4	201.4	208.5
2.456	2.888	2.578	3.015	9.643	2.505	9.95.5	2.448	1.602	3.994	9.40	11.9	5.8	14.2.6	2.369	14.15	3.452	3.47	3.02	2.434	4.053	7.312	3.450	3.623
1.0699	1.9566	1.5476	1.9405	1.9271	1.9686	1.9684	1.0150	0.8698	1.5084	0.5029	0.0	0.0	0.0892	0.4026	0.4508	1.7999	1.6949	1.4980	1.9681	1.2928	0.9586	0.9820	1.9898
1.6719	19.446	1.8949	1.9601	2.1986	1.9970	2.0579	1.5064	1.2367	1.9206	1.0621	0.1458	0.9986	0.8896	1.6627	1.1991	2.5924	2.4482	2.0179	2.1934	1.9780	1.6770	1.6774	1.2006
9.913	4.958	9.969	4.959	9.805	9.885	4.982	9.917	9.870	6.026	1.82.8	1.014	1.787	9.086	4.418	2.784	4.982	4.434	2.689	9.868	6.02	4.915	5.48.1	9.865
1.6719	1.9646	1.8949	1.9601	1.9685	1.9970	1.9579	1.5064	1.2367	1.9206	1.0621	1.1436	1.4035	1.8896	1.1627	1.1991	1.9624	1.4482	1.9079	1.1934	1.9780	1.6770	1.6774	1.2006
7.80	7.7.7	8.0	7.6.9	8.07	7.81	8.2.6	8.11	7.9.4	6.9.0	7.5.9	7.5.7	7.9.8	8.2.8	8.6.1	7.91	7.2.6	7.6.9	7.4.9	7.7.9	8.4.4	6.9.9	8.0.9	8.6.1
2.165	2.169	2.270	2.26.8	2.268	2.168	2.12.4	2.995	2.901	2.269	2.269	2.410	2.419	2.269	2.276	2.210	2.270	2.27.1	2.27.1	2.257	2.2.6.8	2.368	2.2.6.8	2.367
4.97	4.94	9.01	5.05	5.09	4.98	0.05	10.10	15.00	4.86	5.00	4.97	5.00	5.00	4.99	4.96	4.97	5.02	5.00	5.00	4.89	4.98	4.96	4.99
8.09	7.86	7.15	7.89	7.82	6.45	6.84	7.61	7.64	9.26	4.25	4.98	7.46	10.69	15.99	7.92	4.99	4.66	5.68	6.90	7.95	9.84	12.12	6.49
13.00	12.90	12.15	12.89	11.28	11.14	6.10	17.71	27.64	14.12	9.26	9.30	12.66	15.69	16.36	12.50	9.92	9.68	8.68	9.30	11.28	14.81	17.26	11.44
23.16	24.92	23.04	23.19	24.16	24.89	23.86	23.90	26.94	23.49	24.72	23.87	23.87	27.15	27.5.8	23.97	24.33	24.18	24.11	24.85	23.89	23.81	26.36	24.85
5.8	4.7	6.5	6.9	2.1	4.0	5.5	4.2	6.5	6.9	9.7	1.7	18.9	26.8	19.4	7.1	4.0	5.1	4.8	6.4	11.0	6.9	9.7	6.0
8.09	7.41	7.08	7.90	7.67	6.22	6.16	7.81	7.14	9.45	4.96	4.67	7.71	10.14	15.69	7.51	5.04	4.78	5.19	6.42	7.42	9.88	12.22	6.60
0.08	0.05	0.00	0.05	0.09	0.06	0.11	0.16	0.10	0.17	0.12	0.14	0.09	0.09	0.10	0.00	0.08	0.12	0.08	0.12	0.09	0.12	0.10	0.15
18.5	17.4	16.8	18.4	17.8	14.7	17.9	15.2	19.4	21.2	10.6	11.5	18.1	2.9.9	23.5	17.8	12.1	11.4	9.1	18.1	17.5	22.4	27.1	18.4
27.3986	26.084	2.9.6.98	24.1630	2.6.2.87	27.1995	2.6.2.464	27.1990	30.150	2.4.1.249	10.4.9.08	14.9.07	27.1.89.6	1.9.1.3.7.6	2.6.3.0.9.1	1.6.9.8.6.9	2.8.9.6.2.6	3.4.9.6.0.8	2.0.0.6.6.2	4.0.9.6.8.1	4.4.7.1.9.9	2.3.6.4.9.6	3.7.1.8.3.1	4.0.7.4.8.9
2.42.9	2.41.5	2.41.8	2.41.7	2.41.9	2.40.0	2.37.9	2.82.0	2.44.7	2.36.9	2.36.9	2.38.1	2.44.5	2.49.7	2.54.0	2.49.6	2.96.8	2.96.5	2.96.2	2.39.5	2.40.1	2.46.6	2.50.5	2.49.5
2.9	3.0	2.6	2.3	3.8	1.9	2.2	1.9	1.1	9.3	0.6		6.3	1.0	2.1	1.0	2.4	2.2	1.1	2.4	9.8	2.6	9.6	9.0
1.9121	1.86.99	1.6.2.2.56	1.9.0.9.78	2.9.9.80	1.8.8.8.6	1.9.4.7.5.9	1.9.0.9.1.5	1.4.7.9.9	2.5.0.5.7.0	2.9.6.6.6	0.0	0.0	7.9.7.0	1.6.6.2.4	9.2.0.0.8	2.9.2.4.1	2.9.6.5.8	1.9.4.0.6	2.8.1.4.8	2.3.0.9.8	1.6.6.2.0.6	1.8.8.7.2	2.5.7.4.7
1.4.5.4.7.7	1.4.9.6.6.4	1.5.4.8.0.5	1.8.2.9.4.4	1.6.4.6.2.5	1.2.9.9.1.5	1.4.5.7.8.0	1.4.2.2.8.7	1.4.0.9.6.6	1.9.7.5.9.9	1.8.6.8.9.6	2.4.0.5.3	1.5.0.9.2	1.6.1.0.0.8	1.9.6.6.2.8	1.7.4.7.9.2	1.4.5.7.0.6	1.9.6.6.2.8	1.9.4.0.6.6	2.8.1.4.8.8	2.3.0.0.7.9	1.6.6.2.0.6.6	1.8.8.7.7.2	1.8.4.1.6.2	2.5.4.0.6.1
4.9.9.8.7	4.9.9.8.4	4.5.0.4.7	5.0.0.1.0	5.8.4.8.6	3.9.6.5.4	5.4.4.2.5	9.1.2.5.9	1.4.6.5.8	5.1.9.5.0	4.9.2.0	0.0	0.0	3.0.7.7	3.5.8.5.5	1.4.9.2.6	4.8.5.2.1	4.2.1.7.7	1.8.2.1.9	5.7.8.5.0	6.7.2.6.9	5.2.9.3.5	8.9.5.4.6	6.3.9.2.0
3.8.4.7.0.8	3.9.3.9.8.5	2.9.6.7.9.9	3.8.9.9.1.9	4.0.4.1.1.1	2.9.0.0.8.8	3.4.0.4.9.8	2.8.2.1.4.2	2.3.5.6.0.0	4.4.6.2.2.2	1.2.9.0.4.4	2.4.9.2.5	7.5.0.9.0	1.6.8.9.1.5	2.1.9.5.5.0.6	1.1.6.7.9.9.5	2.9.9.3.1.4.4	3.9.8.7.6.4.6	2.1.8.8.1.2.1	4.1.2.8.9.9.6	2.3.9.9.3.5.5	9.0.9.7.8.9.5	8.9.5.4.6.0	6.3.9.2.0.1
3.4.5.4.7.7	1.4.9.6.6.4	1.5.4.8.0.5	1.8.2.9.4.4	1.6.4.6.2.5	1.2.9.9.1.5	1.4.5.7.8.0	1.4.2.2.8.7	1.4.0.9.6.6	1.9.7.5.9.9	1.8.6.8.9.6	2.4.0.5.3	1.5.0.9.2	1.6.1.0.0.8	1.9.6.6.2.8	1.7.4.7.9.2	1.4.5.7.0.6	1.9.6.6.2.8	1.9.4.0.6.6	2.8.1.4.8.8	2.3.0.0.7.9	1.6.6.2.0.6.6	1.8.8.7.7.2	1.8.4.1.6.2	2.5.4.0.6.1
4.9.9.8.7	4.9.9.8.4	4.5.0.4.7	5.0.0.1.0	5.8.4.8.6	3.9.6.5.4	5.4.4.2.5	9.1.2.5.9	1.4.6.5.8	5.1.9.5.0	4.9.2.0	0.0	0.0	3.0.7.7	3.5.8.5.5	1.4.9.2.6	4.8.5.2.1	4.2.1.7.7	1.8.2.1.9	5.7.8.5.0	6.7.2.6.9	5.2.9.3.5	8.9.5.4.6	6.3.9.2.0
3.8.4.7.0.8	3.9.3.9.8.5	2.9.6.7.9.9	3.8.9.9.1.9	4.0.4.1.1.1	2.9.0.0.8.8	3.4.0.4.9.8	2.8.2.1.4.2	2.3.5.6.0.0	4.4.6.2.2.2	1.2.9.0.4.4	2.4.9.2.5	7.5.0.9.0	1.6.8.9.1.5	2.1.9.5.5.0.6	1.1.6.7.9.9.5	2.9.9.3.1.4.4	3.9.8.7.6.4.6	2.1.8.8.1.2.1	4.1.2.8.9.9.6	2.3.9.9.3.5.5	9.0.9.7.8.9.5	8.9.5.4.6.0	6.3.9.2.0.1
1.7.0.9	7.1.2.5	6.6.4.9	7.1.8.2	8.4.6.8	7.4.4.0	7.1.4.2	7.0.0.4	6.1.0.0	7.9.1.9	4.0.8.7	8.8.1	1.5.8.5	2.0.3.5	3.6.1.9	1.2.9	6.4.4.9	8.0.8.5	9.9.9.5	7.0.0.6	6.6.0.7	9.8.2.7	9.6.5.8	0.3.1.0
1.8.7	1.5.1	1.9.9	1.5.6	0.9.9	3.0.8	0.7.7	2.5	6.0.9	0.0.0	0.8.0	0.0.4.2	2.9.4.5	6.9.8	2.2.9	6.7.5	2.4.8	1.5.4	4.4.9	0.4.1	0.0.0	1.8.4	0.4.0	0.4.9
1.7.1	2.1.6	2.2.7	2.5.8	1.9.6	2.1.1	1.8.9	2.2.2	2.5.7	2.4.7	2.5.4	2.5.0	2.2.9	2.4.7	2.2.0	2.1.1	0.4.9	0.6.6	0.8.0	2.1.2	1.7.9	0.9.9	1.4.5	0.9.9
0.5.8.0.0	0.6.1.9.9	0.6.0.1.6	0.6.1.9.0	0.6.0.8.4	0.5.9.8.8	0.6.2.8.0	0.6.8.6.5	0.6.5.4.4	0.6.0.7.5	0.7.8.7.1	0.5.1.6.8	0.6.7.1.4.0	0.6.0.7.6	0.7.6.4.1	0.7.2.1.4	0.9.2.6.5	0.8.8.0.9	0.8.5.9.9	0.6.1.1.5	0.6.0.8.2	0.6.8.7.0	0.6.9.4.0	0.6.1.6.8
19.445	18.671	17.408	18.823	22.184	19.512	18.817	18.585	15.988	20.191	10.879	2.12.5	4.149	7.087	9.9.0.7	9.6.6.4	9.2.9.0.8	9.1.0.1	2.9.0.7.8	2.7.8.9	2.5.4.2.1	1.9.9.7.8	1.9.9.9.8	2.2.8.8.7

(36)	76.9	68.7	STEAM PLUS NON-ENTRAINED VAPOR DISCHARGED FROM VAPOR CONDENSER.
(51)	64.0	90.6	STEAM SHORT CIRCUITED TO CONDENSER [ITEM(9) - ITEM(17)]
(52)	12.9	9.9	NON-ENTRAINED VAPOR FROM SHELL [ITEM(30) - ITEM(51)]

FIG. 3.

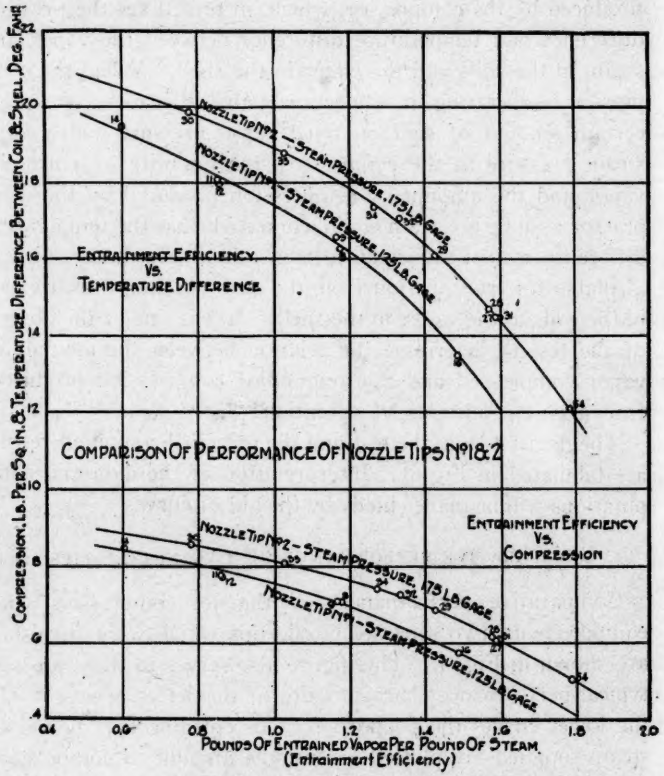
shell depends upon the temperature difference and several operating conditions, such as salinity, water level and condition of heating surface. Further, the quantity of vapor entrained by the compressor depends upon the amount of compression produced by the compressor, which, in turn, fixes the pressure difference and temperature difference between the vapor and steam in the coils and the water in the shell. When the compressor is operating in connection with the evaporator with a certain amount of surface, fixed vapor pressure and a fixed steam pressure at the compressor, the quantity of entrained vapor and the amount of compression produced by the compressor assume a certain equilibrium such that the temperature difference is just sufficient to transmit all the heat which is available for transmission from the steam and entrained vapor in the coils to the water in the shell. It was one of the objects of the test to determine the relation between the amount of vapor compressed and the amount of compression produced, for various conditions of operation.

The data taken in the test and the principal computed results are tabulated in Fig. 3. Interpretation of the data and computations will be made chiefly by the aid of curves.

COMPARATIVE PERFORMANCE OF TWO NOZZLE TIPS.

Comparative performances of the jet compressor when equipped with two separate nozzle tips which were furnished are shown in Fig. 4. This figure also serves to illustrate two typical performance characteristics of the jet compressor. In the lower curves the pounds of vapor entrained per pound of steam supplied are plotted against the amount of compression produced. That there is a definite relation between these two quantities for constant steam and vapor pressures is shown by the fact that, for each nozzle tip, the test points for all runs fall on a smooth curve, which may be termed a characteristic of the compressor. In the upper curves the pounds of vapor entrained per pound of steam are plotted against the temperature differences, and here also the points for each nozzle tip

fall on a smooth characteristic curve. The variation in operating conditions required to cover the range of the curves was obtained by changing either the amount of surface installed or the water level.


Compression, Lb. Per Sq. In. & Temperature Difference Between Coil & Shell Deg. Fahr.
Nozzle Tip No. 2 — Steam Pressure, 175 Lb. Gage
Nozzle Tip No. 1 — Steam Pressure, 175 Lb. Gage
Entrainment Efficiency Vs. Temperature Difference
Comparison Of Performance Of Nozzle Tips No. 1 & 2
Entrainment Efficiency Vs. Compression
Nozzle Tip No. 2 — Steam Pressure, 175 Lb. Gage
Nozzle Tip No. 1 — Steam Pressure, 175 Lb. Gage
Pounds Of Entrained Vapor Per Pound Of Steam (Entrainment Efficiency)

Jet Compressor As Applied To Evaporator

Fig. 4.

It was on account of the better performance of nozzle tip No. 2, as shown by the curves of Fig. 4, that this nozzle tip was installed in the compressor for the main runs. All that

follows pertains to the performance of the compressor with nozzle tip No. 2 installed.

JET COMPRESSOR CHARACTERISTICS.

Two forms of jet compressor characteristics are discussed in a preceding paragraph. Another form of characteristic may be drawn to coördinates of the sum of entrained vapor and steam (coil condensate) vs. temperature difference. Data obtained from the test at various steam pressures and a constant vapor pressure of 5 pounds gage have been plotted on these latter coördinates in Fig. 5. The characteristics were drawn through the test points for gage pressures of 100, 150, 175, 200 and 250 pounds, and the lines for 125 and 225 pounds were put in by interpolation.

ENTRAINMENT EFFICIENCY.

Lines of constant value of the ratio, pounds of vapor entrained per pound of steam used, which might be termed the entrainment efficiency, were drawn by use of the relation:

Weight of condensate equals weight of steam times (1+entrainment efficiency). Even increments of entrainment efficiencies were assumed and the weights of steam were obtained from the calibration runs.

Operation to the left of the line representing an entrainment efficiency of zero is beyond the proper range of operation of the compressor. In this region no vapor is compressed, and part of the steam used by the compressor short-circuits through the vapor line to the distiller condenser. This action takes place in the event of either too small an amount of evaporating surface, or abnormally low water level, as in the case of runs Nos. 44 and 45. Within the range of proper operation of the compressor the entrainment efficiency varied from zero to 1.89 pounds of vapor entrained per pound of steam used.

RELATION BETWEEN THE PRODUCT KS AND PERFORMANCE.

It developed during the test that the performance of the jet compressor in connection with the evaporator was constant for a constant value of the product of heat transfer coefficient and surface. That is, if the compressor operated at a certain point in its characteristic with a value of K=1,000 and S=10 square feet, it would operate at the same point with a value of K=500 and S=20 square feet. By definition

$$K = \frac{\text{B.t.u. per hour}}{S \times T.D} \text{ or } KS = \frac{\text{B.t.u. per hour}}{T.D.}.$$

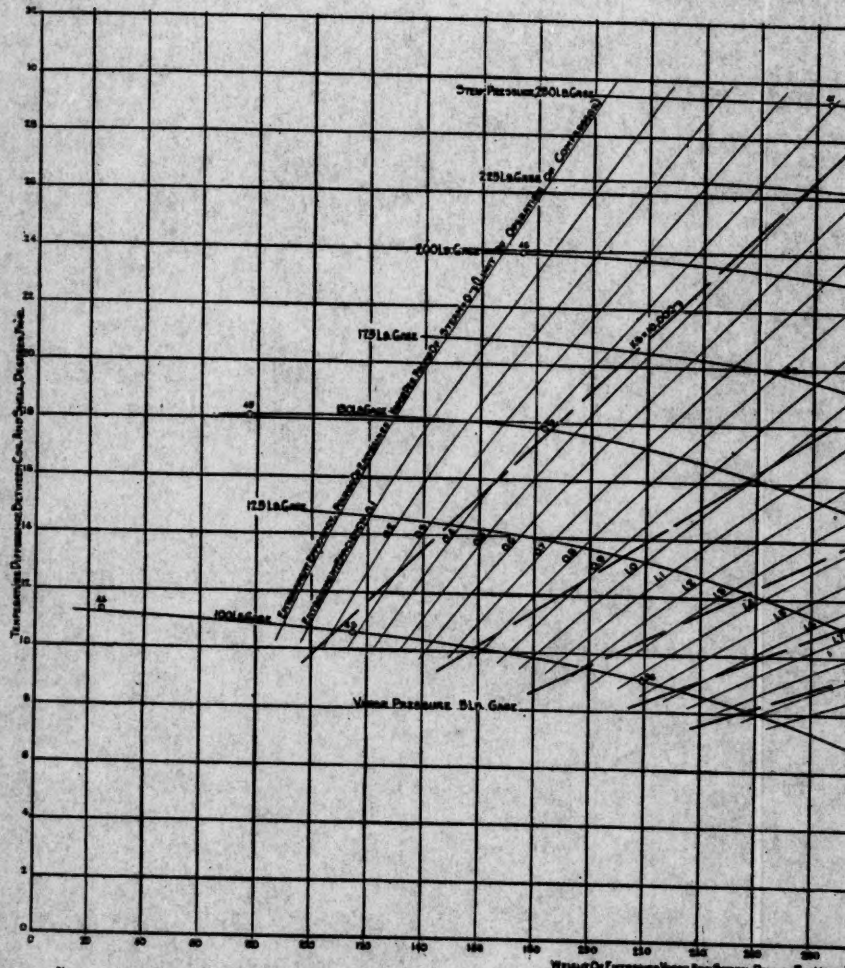
It is evident that each point along the constant steam pressure characteristics of Fig. 5 has one, and only one, value of the quotient:

$$\frac{\text{B.t.u. per hour}}{T.D.} = \frac{\text{B.t.u. per lb.} \times \text{lb. condensate per hr}}{\text{temperature difference}}.$$

In order to illustrate this point fully the curves of Fig. 6 have been drawn to coördinates of weight of condensate vs. the value of the product KS which is measured in B.t.u. per hour per degree temperature difference. The values of the product KS, as determined from the test data tabulated in Fig. 3, are shown by circles. Values of B.t.u. per hour per degree temperature difference, computed from data from the characteristic curves of Fig. 5, are shown as crosses. The agreement between the test points and the computed points fully establishes the relation in question between KS and the quantity of condensate or entrained vapor and steam. Lines of constant value of the product KS have been drawn on the characteristic curves of Fig. 5 by picking off values from the curves of Fig. 6. With a constant amount of surface and a constant value of heat transfer coefficient, operation at variable steam pressure would be along a line of constant KS in Fig. 5.

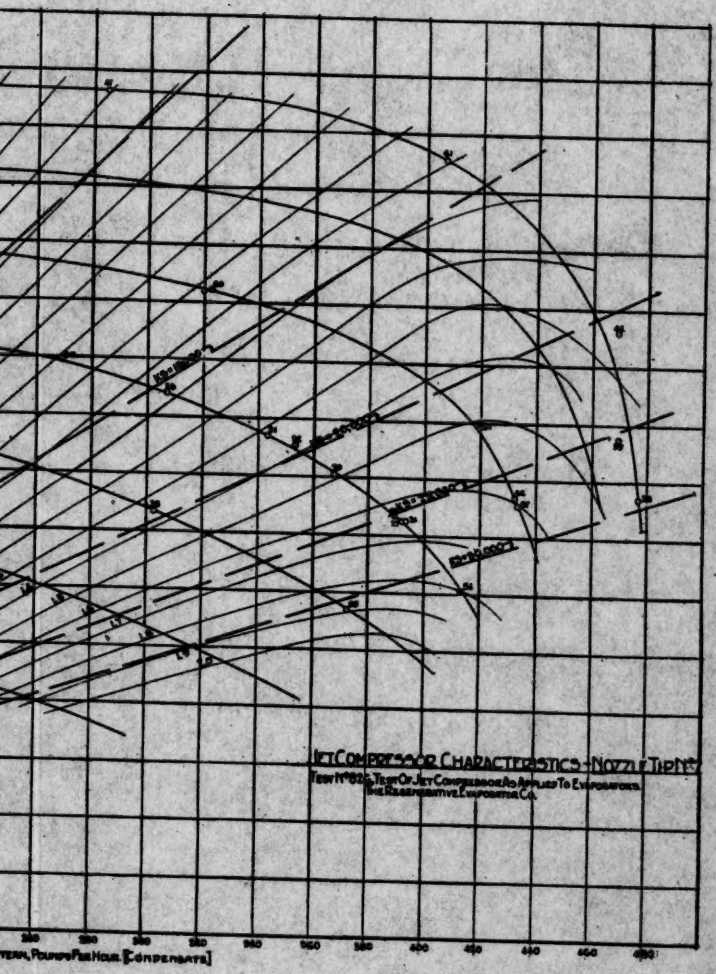
COMPUTATION OF PERFORMANCE.

In Fig. 7 is given a table of performance of the jet compressor when applied to an evaporator with various conditions


Steam Pressure 250 Lbs Gage
225 Lbs Gage
200 Lbs Gage
175 Lbs Gage
150 Lbs Gage
125 Lbs Gage
100 Lbs Gage
Vacuum Pressure 5 Lbs. Gage
Thousands Of Pounds Of Water Evaporated Per Hour From And At 212° Fahr.
Weight Of Entrained Water And Steam, Pounds Per Hour

NOTE:
1 K.D. = Heat Transfer Coefficient Numerically Equivalent To Number Of B.t.u. per Hour, Per Square Foot, Per Degree Temperature Difference.
2. Circles Denote Test Points.

FIG. 5.


JET COMPRESSOR CHARACTERISTICS - NOZZLE TIP N°
TEST N° 82a. TEST OF JET COMPRESSOR AS APPLIED TO EVAPORATORS
THE REFRIGERATIVE EVAPORATOR CO.
POUNDS PER HOUR. [CONDENSATE]

5.

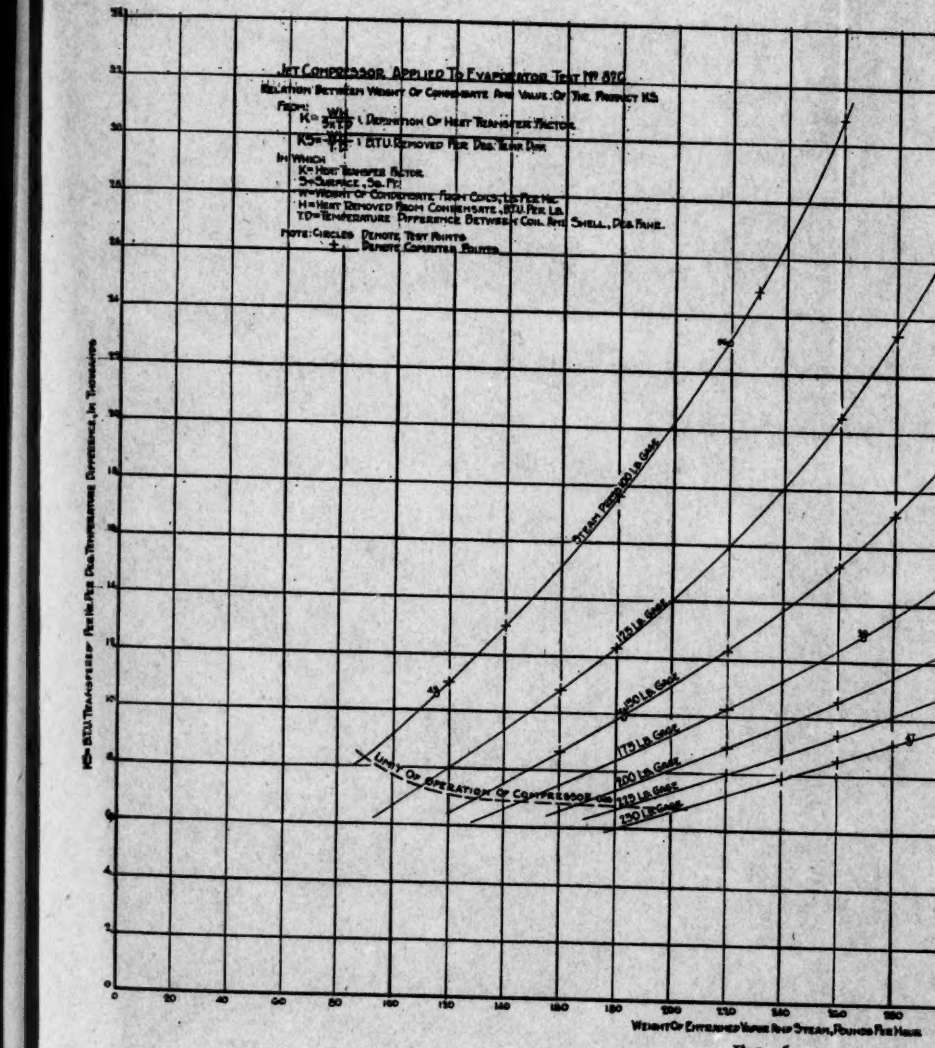
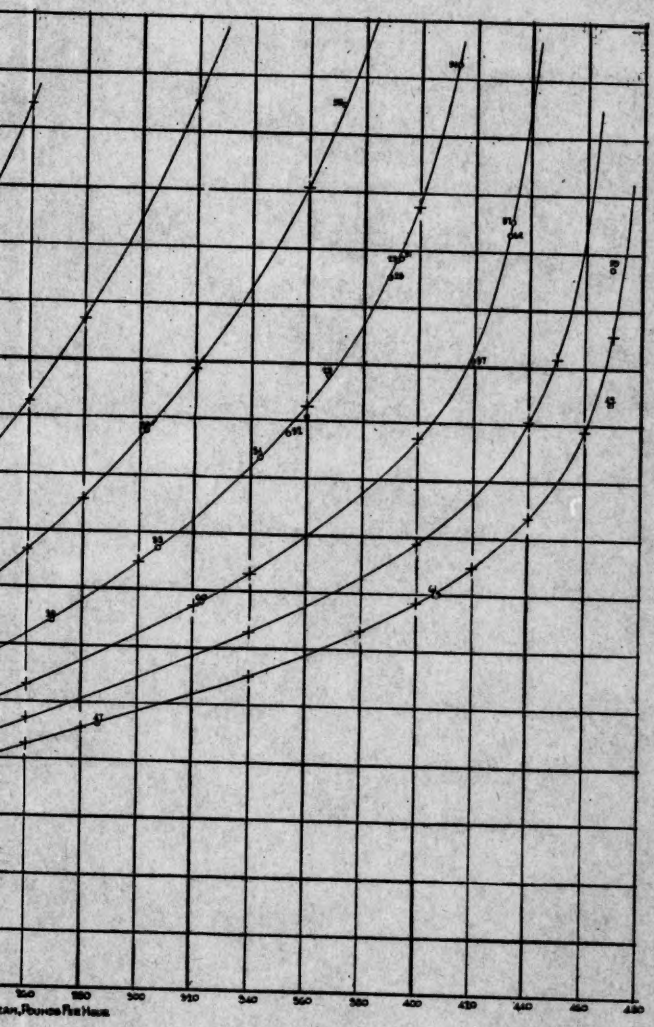

Jet Compressor Applied To Evaporator Test No 870
Relation Between Weight Of Condensate And Value Of The Product K3
From: K= W/ST(T) (Definition Of Heat Transfer Factor.
K3= W/T = 1 B.T.U. Removed Per Deg. Temp Diff.
In Which
K= Heat Transfer Factor.
S= Surface, Sq. Ft.
W= Weight Of Condensate From Coils, Lbs. Per Hr.
H= Heat Removed From Condensate, B.T.U. Per Lb.
T.D.= Temperature Difference Between Coil And Shell, Deg. Fahr.
Note: Circled Points Denote Test Points
Points Connected Plotted
K3= B.T.U. Transferred Per Hr. Per Deg. Temperature Difference, In Thousands
Steam Pressure 240 Lb Gage
175 Lb Gage
150 Lb Gage
125 Lb Gage
100 Lb Gage
Limit Of Operation Of Compressor
Weight Of Entrained Vapor And Steam, Pounds Per Hour

FIG. 6.


Pounds Per Hour

PERFORMANCE OF JET COMPRESSOR AS APPLIED TO EVAPORATOR.

TEST 826.

LINE NUMBER	STEAM PRESS. AT JET COMPRESSOR LB. PER SQ. IN. GAUGE	EVAPORATING SURFACE INSTALLED SQ. FT. WITH K=1000	STEAM USED BY JET COMPRESSOR LB. PER HR.	VAPOR ENTRAINED LB. PER HR.	LB. OF VAPOR ENTRAINED PER LB. OF STEAM USED	COMPRESSION PRODUCED BY COMPRESSOR LB. PER SQ. IN.	PRESSURE IN COIL & LB. PER SQ. IN. GAUGE	TEMP. DIFF. BETWEEN COIL AND SHELL	NON-ENTRAINED VAPOR PRODUCED LB. PER HR.	LB. OF NON-ENTRAINED VAPOR PER LB. OF STEAM USED	TOTAL VAPOR PRODUCED LB. PER HR.	LB. OF TOTAL VAPOR PRODUCED PER LB. OF STEAM	LB. OF STEAM PER LB. OF TOTAL WORK	TOTAL WORK LB. PER HR.	TOTAL VAPOR GAL. PER MIN. PER SQ. FT. OF EVAPORATING SURFACE	SQ. FT. OF EVAPORATING SURFACE REQUIRED PER 1,000 GAL. VAPOR PER MIN. CONDENSED AT 100°
(a)	1	(b)	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
1		10	88.8	78.4	0.1160	4.51	2.91	10.95	570.8	0.8450	649.2	1.0610	0.9421	0.5757	1941	2.874	34.84
2		15	89.8	68.1	0.7699	4.06	9.06	10.12	77.9	0.8675	147.0	1.6970	0.6609	1.133	2.828	1.5350
3	100	20	89.8	108.4	1.2071	3.85	6.82	9.92	74.1	0.8251	182.5	2.0323	0.4921	2.123	3.643	3.613
4		25	89.8	141.2	1.5724	3.57	8.52	9.62	71.1	0.7918	212.3	2.3641	0.4230	2.603	3.464	4.037
5		30	89.8	166.4	1.8530	3.25	8.18	8.16	60.0	0.7684	235.4	2.6214	0.3815	3.017	3.264	4.434
6		10	108.9	98.6	0.7635	6.00	11.00	14.40	93.0	0.9041	158.8	1.5678	0.7902	1.238	3.963	2.510
7		15	108.9	101.4	0.9860	5.54	10.54	13.40	91.2	0.8425	192.7	1.7689	0.8609	1.521	3.972	2.694
8	125	20	108.9	166.6	1.3868	5.01	10.01	12.22	88.2	0.8098	234.8	2.1443	0.6004	1.781	3.816	3.263
9		25	108.9	180.3	1.6460	4.45	9.46	11.03	88.3	0.7790	268.6	2.4196	0.4125	2.143	3.500	3.779
10		30	108.9	208.9	1.8767	4.09	9.09	10.04	89.1	0.7899	298.6	2.6598	0.3794	2.338	3.277	4.016
11		10	128.8	96.2	0.8208	7.60	12.66	17.77	113.4	0.8850	169.6	1.3960	0.7894	1.481	4.018	2.559
12		15	128.8	127.2	0.9876	6.86	11.86	16.24	107.1	0.8515	234.3	1.8191	0.8487	1.843	4.149	2.727
13	150	20	128.8	177.2	1.3756	6.07	11.07	14.57	101.4	0.7917	278.5	2.1609	0.6491	2.287	4.021	2.891
14		25	128.8	240.0	1.6615	5.36	10.36	13.00	99.1	0.7694	339.1	2.6309	0.4114	2.901	3.960	3.176
15		30	128.8	259.5	1.8636	4.78	9.75	11.65	97.1	0.7547	397.1	2.6172	0.3841	3.401	3.921	3.036
16		10	147.8	67.2	0.4547	8.56	13.36	20.35	136.0	0.8865	198.5	1.3410	0.7487	1.926	4.230	2.577
17		15	147.8	153.2	1.0368	8.19	13.19	18.97	121.9	0.8860	274.1	1.8549	0.5392	2.662	4.216	2.803
18	175	20	147.8	208.2	1.4087	7.13	12.13	16.80	115.6	0.7821	323.8	2.1908	0.4565	3.145	4.166	2.990
19		25	147.8	241.2	1.6369	6.15	11.15	14.75	113.4	0.7675	394.6	2.6998	0.4168	3.804	4.108	3.693
20		30	147.8	299.9	1.7987	5.55	10.55	12.95	112.6	0.7616	371.5	2.9200	0.3966	4.019	3.967	3.804
21		10	166.6	78.4	0.4756	10.48	15.48	23.91	145.7	0.8882	216.1	1.3171	0.7768	2.004	4.631	2.598
22		15	166.6	180.2	1.0816	9.38	14.38	21.75	134.0	0.8045	244.2	1.8860	0.5302	2.903	4.603	2.661
23	200	20	166.6	238.3	1.4304	8.33	13.32	19.05	128.9	0.7737	367.2	2.2041	0.4537	3.547	4.518	2.998
24		25	166.6	261.8	1.5714	6.89	11.89	16.30	128.1	0.7688	389.9	2.3405	0.4273	3.770	4.448	3.134
25		30	166.6	271.8	1.6313	5.78	10.78	13.95	125.8	0.7553	400.6	2.4048	0.4159	3.918	4.364	3.209
26		10	185.4	84.4	0.4550	12.01	17.01	26.35	160.0	0.8632	239.4	1.2908	0.7746	2.188	4.539	2.673
27		15	185.4	203.6	1.0977	10.92	16.91	24.32	147.6	0.7961	321.0	1.7312	0.5775	3.001	4.675	2.857
28	225	20	185.4	254.3	1.3716	9.04	14.04	20.67	143.8	0.7758	398.1	2.1473	0.4657	3.656	4.575	2.960
29		25	185.4	272.6	1.4699	7.58	12.58	17.92	144.3	0.7778	416.6	2.2470	0.4450	3.818	4.529	3.088
30		30	185.4	278.6	1.5026	6.30	11.30	14.85	144.9	0.7814	423.7	2.2851	0.4376	3.883	4.497	3.149
31		10	204.0	105.0	0.5147	13.67	18.65	29.25	176.3	0.8647	281.3	1.3794	0.7249	2.608	4.809	2.737
32		15	204.0	216.9	1.0631	11.96	16.96	26.13	164.4	0.7861	373.9	1.8899	0.5578	3.494	4.778	2.876
33	250	20	204.0	297.1	1.3503	9.97	14.97	21.72	156.6	0.7525	417.7	2.0478	0.4884	3.936	4.681	3.046
34		25	204.0	309.6	1.5486	7.71	12.71	18.00	165.1	0.7946	474.1	2.2795	0.4390	4.040	4.586	3.217
35		30	204.0	273.8	1.5433	6.06	11.00	14.40	164.8	0.8098	478.2	2.1480	0.4659	4.214	4.512	3.291

NOTES:
1 THE VALUES TABULATED ARE FOR THE FOLLOWING OPERATING CONDITIONS.
(a) Pressure Of Vapor In Shell, 5 lb Gage
(b) Water Level Such As To Submerge All Heating Surface.
(c) Feed-Water Heated To 200 Deg. Fahr.
(d) Blow-Down Or Radiation Loss = 5 %.
(e) Average Salinity Of Water In Shell = $\frac{5}{32}$ Grains.
(f) Heat Transfer Coefficient, K = 1000 B.T.U. Per Sq. Ft. Per Hr. Per Degree Temp. Difference.
2 For Heat Transfer Coefficient, K, Other Than 1000, Columns (2), (5) & (6) Will Be Changed Inversely With The Value Of K, & Columns (15) Will Be Changed Directly With Value Of K.

Fig. 7.

of steam pressure and heating surface. The method of computing the performance from the test data is briefly as follows: Assuming various values of steam pressure and of the product KS, the weight of entrained vapor and steam (condensate) is determined from the curves of Fig. 5. It may be noted that col. 2 in the table of performance Fig. 7, which is headed "Evaporating Surface Installed, square feet, with K=1,000," may also be considered as representing the product of KS, expressed in thousands. The amount of steam used is determined from the calibration of the steam jets of the compressor. The temperature difference is obtained from the characteristic curves of Fig. 5, corresponding to the steam pressure and the amount of condensate. The amount of compression is obtained from steam tables, corresponding to the vapor pressure and to the temperature difference. The amount of entrained vapor is the amount of condensate minus the amount of steam used. The amount of non-entrained vapor produced is obtained from the heat balance, the equation for which is:

$$W_sH_s + W_eH_e = W_eH_v + W_nH_v + W_eH_w + H_r,$$

in which

W_s = Weight of steam, pounds per hour.

W_e = Weight of entrained vapor, pounds per hour.

W_n = Weight of non-entrained vapor, pounds per hour.

H_s = Heat in the steam, above drain temperature B.t.u. per pound.

H_e = Heat in the entrained vapor, above drain temperature B.t.u. per pound.

H_v = Heat in the vapor, above feed temperature B.t.u. per pound.

H_w = Heat equivalent to the work of adiabatic compression of the entrained vapor, from vapor pressure to coil pressure, B.t.u. per pound.

H_r = Heat lost in radiation and blow down, B.t.u. per hour assumed to be 5 per cent of the total amount of heat supplied to the coils per hour.

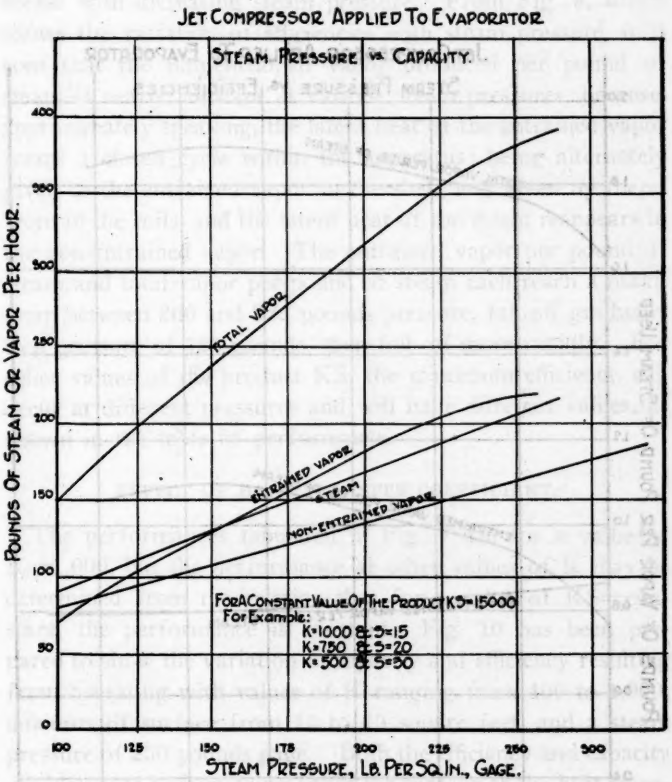


Fig. 8.

EFFECT OF STEAM PRESSURE ON CAPACITY AND EFFICIENCY.

In order to show graphically certain features of the performance, several curves plotted from the data given in the table of performance have been prepared and will be discussed in turn.

Fig. 8 shows that for a value of the product KS=15,000, the steam, non-entrained vapor, and entrained vapor each in-

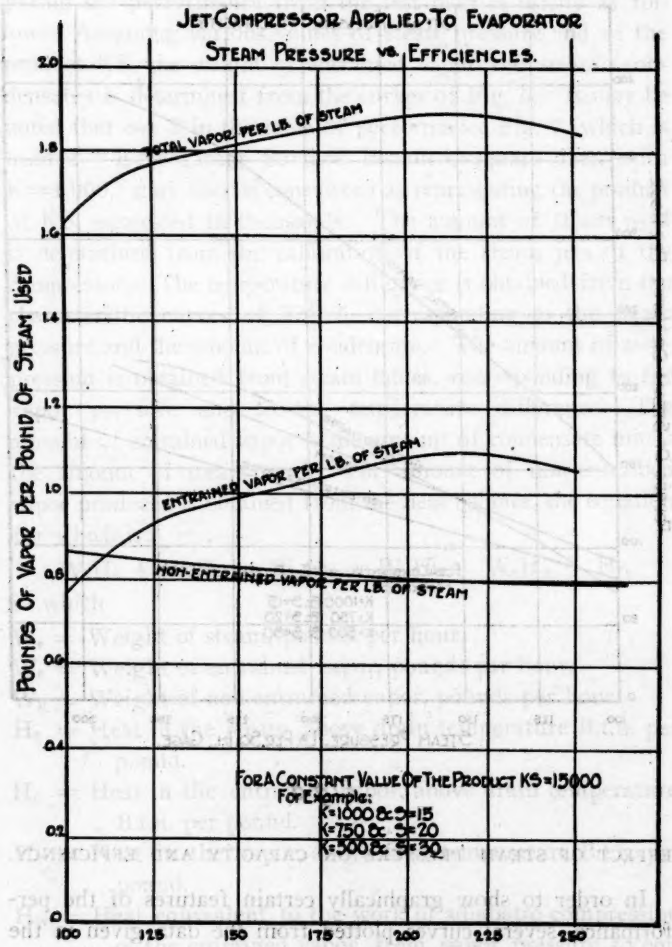

JET COMPRESSOR APPLIED TO EVAPORATOR
STEAM PRESSURE VS. EFFICIENCIES.
TOTAL VAPOR PER LB. OF STEAM
ENTRAINED VAPOR PER LB. OF STEAM
NON-ENTRAINED VAPOR PER LB. OF STEAM
POUNDS OF VAPOR PER POUND OF STEAM USED
2.0
1.8
1.6
1.4
1.2
1.0
0.8
0.6
0.4
0.2
0
100
125
150
175
200
225
250
FOR A CONSTANT VALUE OF THE PRODUCT KS=15000
FOR EXAMPLE:
K=1000 & S=15
K=750 & S=20
K=500 & S=30
STEAM PRESSURE, LB. PER SQ. IN., GAGE.

Fig. 9.

crease with increasing steam pressure. From Fig. 9, which shows the variation of efficiencies with steam pressure, it is seen that the non-entrained vapor produced per pound of steam is nearly constant at various steam pressures, because, approximately speaking, the latent heat of the entrained vapor forms a closed cycle within the apparatus; being alternately given to the entrained vapor in the shell, and given up therefrom in the coils, and the latent heat of the steam reappears in the non-entrained vapor. The entrained vapor per pound of steam and total vapor per pound of steam each reach a maximum between 200 and 225 pounds pressure, fall off gradually to a pressure of 125 pounds, then fall off more rapidly. For other values of the product KS, the maximum efficiency will occur at different pressures and will have different values, as shown in the table of performance.

EFFECT OF HEAT TRANSFER COEFFICIENT.

The performances tabulated in Fig. 7 are for a value of K=1,000, but the performance at other values of K may be determined from the relation that for a value of KS=constant, the performance is constant. Fig. 10 has been prepared to show the variation in capacity and efficiency resulting from operating with values of K ranging from 400 to 1,000, amounts of surface from 15 to 40 square feet, and a steam pressure of 250 pounds gage. Both the efficiency and capacity are increased by increasing either the surface or the heat transfer coefficient. For the smaller amounts of surface there is a considerable variation in capacity and efficiency with heat transfer coefficient, but with the larger amounts of surface installed there may be a considerable variation in the value of the heat transfer coefficient with little effect upon capacity and efficiency.

A diagrammatic representation of the action and performance of the regenerative evaporator is given in Fig. 11. The quantities are taken from the data represented by line (18) in the table of performance, Fig. 7, for a steam pressure of 175

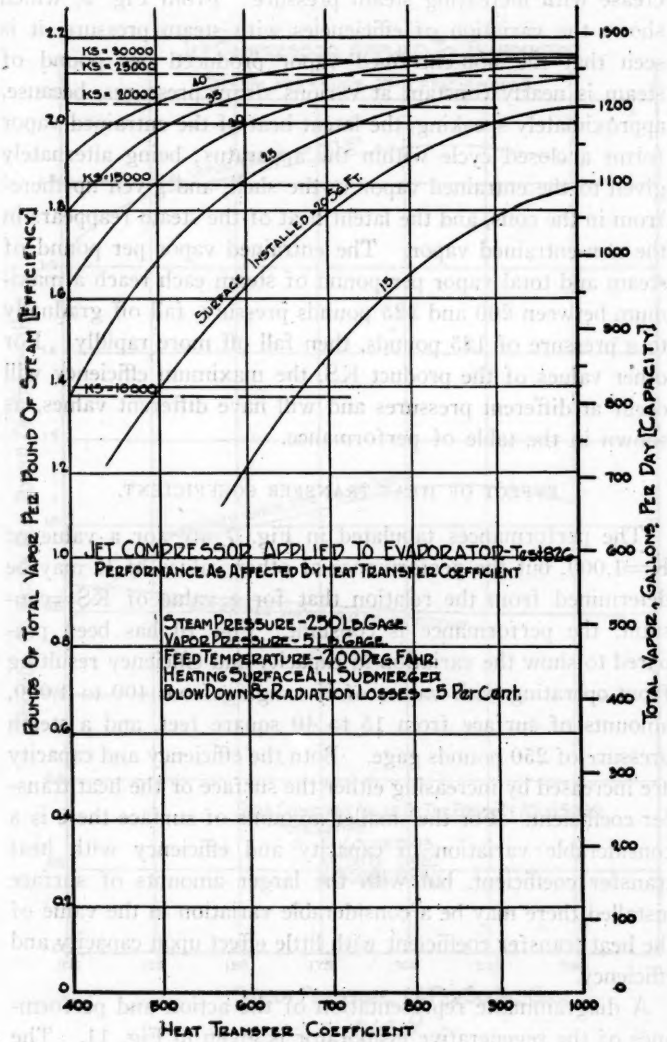

JET COMPRESSOR APPLIED TO EVAPORATOR-Test 826
PERFORMANCE AS AFFECTED BY HEAT TRANSFER COEFFICIENT
STEAM PRESSURE - 250 LB GAGE.
VAPOR PRESSURE - 5 LB GAGE.
FEED TEMPERATURE - 200 DEG. FAHR.
HEATING SURFACE ALL SUBMERGED
BLOW DOWN & RADIATION LOSSES - 5 PER CENT.
KS = 30000
KS = 25000
KS = 20000
KS = 15000
KS = 10000
40
35
30
25
20
15
SURFACE INSTALLED - 40 Sq. Ft.
POUNDS OF TOTAL VAPOR PER POUND OF STEAM [EFFICIENCY]
TOTAL VAPOR, GALLONS PER DAY [CAPACITY]
HEAT TRANSFER COEFFICIENT
2.2
2.0
1.8
1.6
1.4
1.2
1.0
0.8
0.6
0.4
0.2
0
1500
1200
1100
1000
900
800
700
600
500
400
300
200
100
0
400
500
600
700
800
900
1000

Fig. 10.

pounds gage and a value of the product KS equal to 20,000, that is, 20 square feet of surface with a heat transfer coefficient of 1,000. This diagram shows the manner in which 323.8 pounds of vapor per hour is produced with 147.8 pounds of steam per hour.

The heat balance of the system is shown diagrammatically in Fig. 12, which gives the distribution of all the heat units involved in the production of one pound of vapor under the same conditions as apply to Fig. 11. This sketch also serves to explain the action of the regenerative system. 1.10 pounds

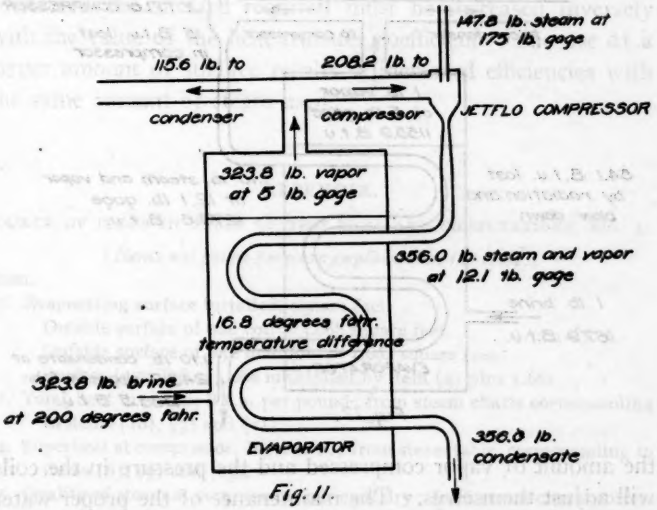


Fig. 11.

of a mixture of steam and vapor is required in the coils to produce 1 pound of vapor, in this respect the action being similar to the usual single-effect evaporator. The efficiency of the regenerative system lies in the fact that of the 1.10 pounds of mixture required in the coils only 0.46 pound is required from the steam line and the remainder, 0.64 pound is obtained from the entrained vapor which is compressed and returned to the coils and condensed, in place of going to the distiller condenser to be condensed.

OPERATION.

During operation of the evaporator the main steam valve is opened wide, no throttling being required unless it is desired to operate at reduced capacity. When operating in connection with a distiller condenser in which atmospheric pressure is carried, the vapor valve to the condenser is opened wide. The valve in the vapor line to the compressor is left open wide, and

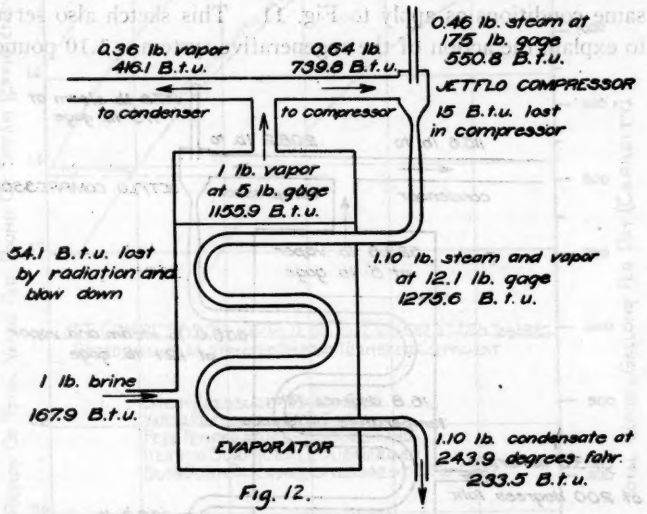


Fig. 12.

the amount of vapor compressed and the pressure in the coils will adjust themselves. The maintenance of the proper water level, bleeding of air from the coils, attention to the drain from the coils and the maintenance of the proper salinity of the brine must be taken care of as in ordinary evaporator operation.

SUMMARY.

The system for the regenerative operation of evaporators, whereby part of the vapor is compressed by means of a jet compressor and returned to the coils, produces approximately the efficiency of a triple-effect plant in a single evaporator

shell. The system operates with a relatively low temperature difference, and the amount of surface required to produce the desired capacity must be proportioned in accordance with the values tabulated in Fig. 7. An economical point for the design and operation of the system tested is for a steam pressure of 225 pounds gage and 17.48 square feet of surface per 1,000 gallons per day, assuming a heat transfer coefficient of 1,000. For these conditions the temperature difference will be 20 degrees F., and 2.15 pounds of total vapor is produced per pound of steam used. For lower values of heat transfer coefficient the surface required must be increased inversely with the value of the heat transfer coefficient. The use of a larger amount of surface results in increased efficiencies with the same amount of steam used.

APPENDIX.

SOURCE OF ITEMS IN TABLE OF TEST DATA AND COMPUTATIONS, FIG. 3.

(*Items not given here are explained in the table.*)

Item.

7. Evaporating surface installed, square feet.
   Outside surface of one coil = 3.068 square feet.
   Outside surface of one manifold = 1.662 square feet.
   Surface installed = 3.068 multiplied by item (4) plus 1.662.
12. Total heat in steam, B.t.u. per pound; from steam charts corresponding to items (10), (9) and (11).
14. Superheat at compressor, degrees F.; from steam table, corresponding to items (8), (9) and (13).
15. Quality of steam at compressor, per cent dry; from steam charts corresponding to items (12), (8) and (9).
16. Quality at 250 pounds gage, per cent dry; from steam charts corresponding to item (12) and steam pressure of 264.7 pounds absolute.
19. Steam used by compressor, pounds per hour; from calibration curve of nozzle tip, corresponding to absolute pressure of steam at compressor. (Steam flow through the nozzle depends only on inlet steam pressure when the absolute back pressure is less than 0.58 of the steam pressure.)
28. Vapor pressure, gage, pounds per square inch; from readings of mercury manometer corrected for water which collected over mercury column.
29. Pressure difference between coil and shell, pounds per square inch; from reading of differential mercury manometer, corrected for water which collected over mercury columns.

32. Superheat at coil entrance, degrees F.; from steam tables corresponding to items (30), (9) and (31).

33. Compression produced by compressor, pounds per square inch; from reading of manometer corrected for water which collected over mercury columns.

35. Temperature difference between coil and shell, degrees F. Temperature corresponding to absolute coil pressure minus temperature corresponding to absolute vapor pressure.

36. Heat given to vapor, B.t.u. per hour $(H_v - q_f)\ W_v$.

In which

H_v = Heat of vapor above 32 degrees B.t.u. per pound; from steam tables corresponding to items (28) and (9).

q_f = Heat of the liquid of the feed, B.t.u. per pound; from steam tables corresponding to feed temperature item (26).

W_v = Weight of total vapor, pounds per hour, item (21).

38. Sub-cooling of drain, degrees F. Temperature corresponding to item (30) and (9) minus item (37).

39. Heat of entrained vapor above drain temperature, B.t.u. per hour.

$(H_v - q_d)\ W_e$.

In which

H_v = Heat in the vapor above 32 degrees, B.t.u. per pound, corresponding to item (28) and (9).

q_d = Heat of the liquid of the drain B.t.u. per pound, corresponding to item (37).

W_e = Entrained vapor, pounds per hour, item (20).

40. Heat of steam above drain temperature, B.t.u. per hour.

$(H_s - q_d)\ W_s$.

In which

H_s = Item (12).

q_d = Heat of the liquid of the drain B.t.u. per pound, as in item (39).

W_s = Item (19).

41. Heat equivalent of the work of compression of entrained vapor, B.t.u. per hour.

$(H_e - H_v)\ W_e$.

In which

H_v = Same as in item (39).

H_e = Heat in the vapor B.t.u. per pound after adiabatic compression from dry saturated steam corresponding to item (28) and (9) to the pressure corresponding, item (20) and (9).

W_e = Same as in item (39).

45. Heat transferred per square foot of evaporating surface, per degree T.D., B.t.u. per hour.

$$\frac{\text{Item (43)}}{\text{Item (7)} \times \text{Item (35)}}.$$

49. KS = B.t.u. per hour per degree T.D.; from the formula

$$K = \frac{\text{B.t.u. per hour}}{\text{T.D.} \times S}.$$

$$KS = \frac{\text{B.t.u. per hour}}{\text{T.D.}} = \text{Item (45)} \times \text{item (7) or item (43)} \div \text{item (35)}.$$

In which

K = Item (45).

S = Item (7).

B.t.u. per hour = item (43).

T.D. = Item (35).

NOTE.—All thermometer readings were corrected for calibration error and exposed stem. All gage readings were corrected for calibration error and water in syphon.

Marks and Davis' Steam Tables and Ellenwood's Steam Charts, which are based on Marks and Davis' Tables, were used.

LATERAL FLOW OF WATER UNDER LOW HEADS.

By Albert E. Guy, Associate.

In practice the velocity of flow of water in piping and conduits ranges from five to eighteen feet per second, and the heads generating these velocities may be said to range roughly from ⅔ to 5 feet. The pressures utilized in pumping systems vary from a few feet to several hundred, depending, of course, upon the kind of application and the location of the plant. But even with a high-pressure system the conditions of flow within it—that is, from one given part to one adjacent—depend almost entirely upon a flow head of a few feet. Failure to realize and appreciate that fact has proven in many instances very detrimental to the effectiveness of hydraulic installations.

Since the frictional losses vary nearly as the square of the velocity of flow, the latter should be made very small; on the other hand, a small velocity entails the use of piping of large diameter, with correspondingly large fittings and valves, all of which make the first cost undesirably high. Sufficient information is at hand, however, to enable the designer to determine the velocity and sizes most appropriate to insure a minimum of loss at the least cost of material, based usually upon operation through a given number of years. It is therefore practical observation which has determined the velocity limits stated above.

This paper is intended mainly to deal with the question of head causing the flow through a lateral orifice. This question has been studied at great lengths and debated by many authorities. Admittedly it has not been settled, but the point to be exposed here seems to be novel, for one thing, and then, also, it is of great practical interest.

THE VELOCITY HEAD.

A container, filled with water, is provided with a lateral orifice near its lower end. Allowing the water to flow freely out of the orifice into the atmosphere, it is admitted that the velocity of flow through the most constricted section of the issuing jet is generated, at a given instant, by a *head* or column of water having for height the distance from the center of the orifice to the level of the water in the container at that instant. This was brought to light as a theorem by Toricelli, in 1643.

Friction against the orifice's walls diminishes somewhat the velocity; apart from that, authorities seem agreed that the head thus measured represents but an approximate average between those obtaining at the upper and lower strata of the jet. In other words, calling H the distance from level to center of orifice, and d the vertical thickness of the jet, H is accepted as the average head between $\left(H+\frac{d}{2}\right)$ and $\left(H-\frac{d}{2}\right)$, when the head is large relatively to the vertical height of the orifice.

As the head is made smaller, the condition approaches gradually that of the flow in open conduits or in river beds. It seems, then, evident that Toricelli's theorem must be modified to suit the case. The early hydraulicians must have paid much attention to that proposition, as is evidenced by its treament in Bossut's Hydraulique (Paris, 1786), in Chapter 3, p. 265. This eminent author gives what appears to have been the accepted demonstration at that time. Assume, said he, that the orifice is entirely closed by a thin plate firmly fixed in place, no flow can issue. Now, let a number of small holes be cleanly drilled in the plate along the vertical axis of the orifice. From each hole a jet of water will issue, and, assuming also that the friction is negligible, the jets will issue with increasing force as the holes get nearer the lower edge of the orifice. The Toricelli proposition holds strictly true in the case of a very small orifice, and the head producing the flow is correctly

represented by the vertical height measured from the center of the orifice to the level of the water in the container. Therefore, there will be as many different heads as there are holes vertically aligned at the orifice. Proceeding further, if the plate is now perforated with a very large number of such small holes, we may conceive that, at the limit, the volume issuing from the totality of the holes will equal that which would issue freely through the unhindered orifice. It follows that, in order to calculate the volume issuing from a given orifice in the unit of time, either the effective mean velocity must be determined, or some integrating process must be found. That is not difficult; for instance, Fig. 1 represents a container provided with a rectangular orifice $a \times b$, and the correct head for a very thin horizontal stratum dh is h. The velocity per second is $v = \sqrt{2gh}$, and the volume issuing is $v \times b \times dh = b.\sqrt{2gh}.dh = d\theta$. The total volume for the whole orifice is evidently $\theta = b\sqrt{2g}\int_{H-a}^{H} h^{\frac{1}{2}}dh = \frac{2}{3}b\sqrt{2g}\left[H^{\frac{3}{2}} - (H-a)^{\frac{3}{2}}\right]$. It is, of course, well understood that, because of the phenomenon of contraction of the jet issuing from the orifice, only about $\frac{5}{8}$ of the volume thus calculated passes actually through the orifice.

Bossut himself was in doubt as to the correctness of the above reasoning, arguing that, so long as the holes in the thin plate were few and their total area of passage small relatively to the full-sized orifice, the volume issued from each hole depended upon the velocity due to the height of water above the center of the hole. But with an infinite number of small holes it was not clear that the total mass of the streams was to issue in the same manner as from isolated small holes. However, since the hypothesis gave results conforming very closely with those of experience he accepted it as a basis for the solution of numerous problems.

The same formula is given in most treatises on Hydraulics; it is, however, modified by the adjunction of a coëfficient of discharge, differing little from $\frac{5}{8}$, which takes care of the

contraction effect and of some friction along the edges of the orifice.

If the hypothesis holds for an orifice located much below the level in the container, as in Fig. 1, it should still better apply to an orifice partly filled (vertically) by the outflowing water. Such a case is represented in Fig. 2, copied from the published Experiments of Poncelet and Lesbros (Paris, 1832). The outlines of the vein were measured with minutious care

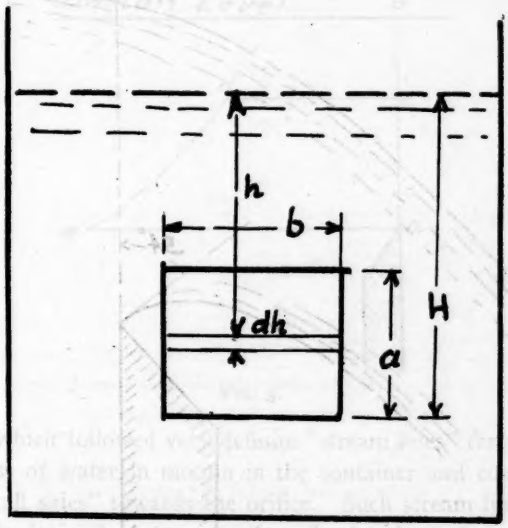


FIG. 1.

by these skilled experimenters. The opening was a square of 7⅞ inches per side. The plan view is not shown here, but cross sections of the vein at various points from the orifice are given in the original work, and they indicate the usual lateral convergence observed in similar cases. At some distance from the orifice the vein attains its maximum contraction, which, however, it is not possible to locate exactly, because then the velocity of flow generated by the effective upstream head is increased through the action of gravity upon the whole vein

past the orifice. This latter action is most strongly apparent when the initial velocity is small, as in this case.

One most important point must be recorded at once, that is the very vigorous rise of the vein past the lower edge of the orifice, the angle at the edge being 54 degrees. Now, according

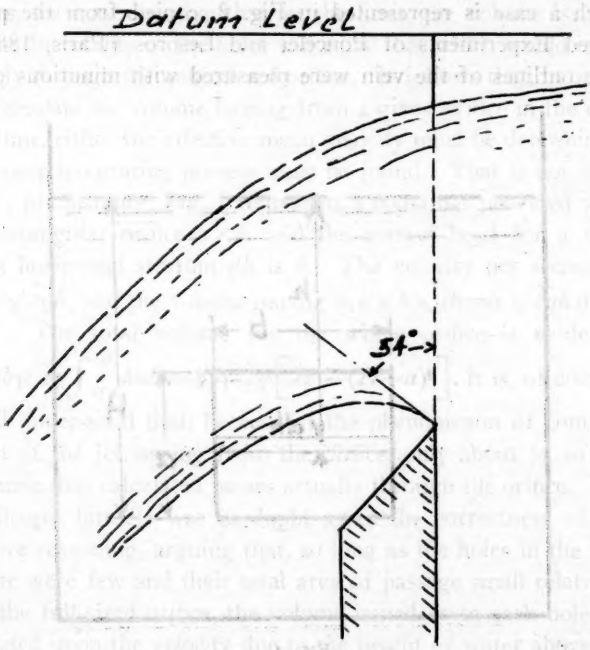


FIG. 2.

to the theory explained by Bossut, the flow head should be the greatest above the edge. Consequently, the velocity would be maximum there, and the underside of the vein would be expected to remain a horizontal plane for quite a distance from the sill. Incidentally, the velocity at *c*, the top of the vein at orifice, Fig. 3, being represented by a length *ed*, corresponding to the head *bc;* that at *a* represented by *ac*, due to head *ab*, for any given head along *ab* the velocity would be measured by a horizontal line extending from *ab* to curve *de*.

The rise of the lower part of the vein may be observed with any kind of orifice when the depth of water from the sill to the bottom of the container is larger than the vertical thickness of the vein through the orifice. *It does not appear that the true significance of such rise has ever been realized,* although, in many instances, observers have noted and plotted the paths of chemicals or other substances dropped in the upstream

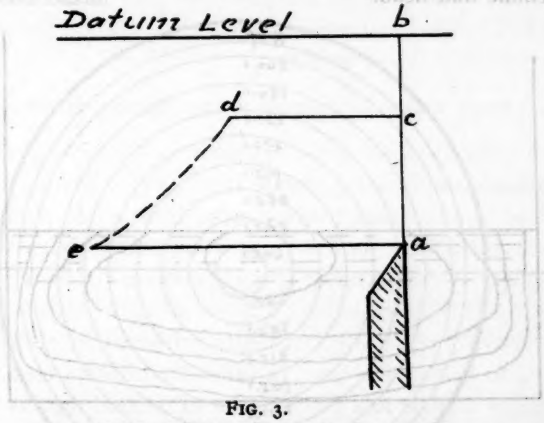


FIG. 3.

water, which followed very definite "stream lines" throughout the mass of water in motion in the container and converged " from all sides" towards the orifice. Such stream lines rose distinctly from the bottom to the sill. In view of such clear observations the " varying flow-head theory" seems untenable.

Bossut, in reviewing the efforts of earlier investigators, cites Newton as the first real observer of the effects of vein contraction. But while taking it into account, in the 1714 edition of the Principia, Newton did not abandon his theory of flow, based upon the formation of a " cataract" within the mass of liquid in the container, whereby only a certain portion of the mass " fell" constantly from the upper level directly toward the orifice. This formation, says Bossut, is contrary to the laws of Hydrostatics and to experience, which unite in show-

ing that when water flows from the orifice of a container, *all particles move towards the orifice*. This conclusion is eminently sound, and now, since it is so well known that all the particles in the container do move in a general direction, *it is obvious that the motion of any given set of particles forming an elementary stream must be caused by a force, or head, affecting that particular stream.* The problem is, then, to determine that head.

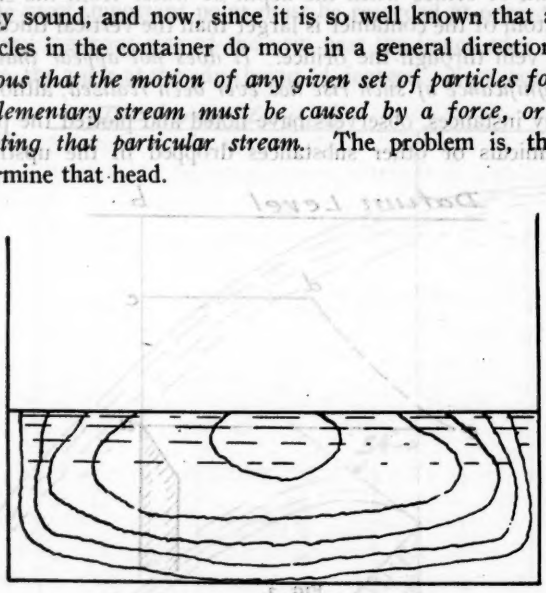


FIG. 4.—LINES OF EQUAL VELOCITIES.
[From Darcy]

PATHS OF EQUAL VELOCITY.

Passing now to observations of a different order—that is, *on the general distribution of velocity at different points in the cross-section of a channel,* another most important point must also be recorded. Thus, Fig. 4, reproduced in part from Unwin's Hydraulics, page 253, represents the cross section of a rectangular channel 0.8 meter wide; the velocity of the water was observed at 36 different points, and the curves of equal velocity were plotted, as shown, by Darcy, in 1857. Similarly, Fig. 5, reproduced from page 219 of the same book, shows the cross section of a stream flowing through a cement pipe of 0.8 meter in diameter. Gagings were taken by Bazin,

in 1897, and each of the concentric circles is very nearly a curve of equal velocity. The figures are the values of the different velocities recorded along the vertical diameter of the section. Bazin experimented also with pipes of rectangular cross section and found that the paths of equal velocity were practically concentric rectangles. A number of other reliable observers have found results substantially similar to those of Darcy and Bazin.

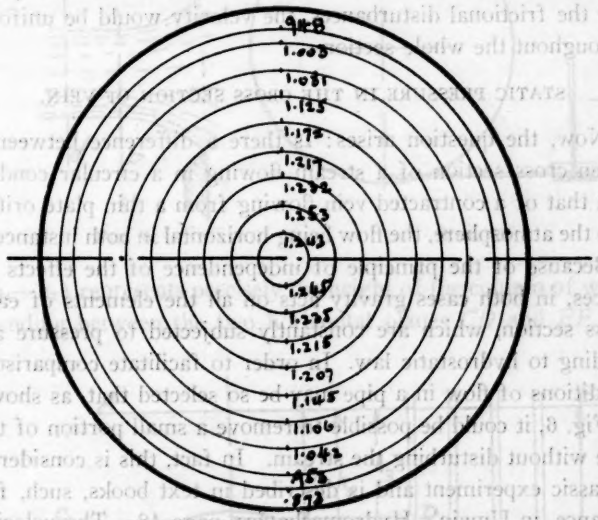


FIG. 5.

Referring again to Fig. 5, the predominant fact is certainly that the velocity is very nearly constant along one given circle, and, in particular, *that it is practically the same at the lower extremity of a vertical diameter as at the upper end.* It is not difficult to realize that the concentric layers of water nearest the wall of the pipe are affected by friction and that, consequently, this effect being gradually transmitted to the whole mass, the velocity of the layers increases from the wall towards the axis. It might also be admitted that the velocities in the lower half of the section are a little smaller than the

corresponding ones in the upper half, because the hydrostatic pressure causes the flowing mass to adhere closely to the lower surface of the pipe, while the disengagement of air at the top may reduce the intensity of contact thereabout, and, consequently, that of friction. It seems evident that (1) *since at the two ends of a vertical diameter the velocity is the same, that velocity is due to one head only,* which, in this case, corresponds to the slope of the pipe; and (2) that, were it not for the frictional disturbances, the velocity would be uniform throughout the whole section.

STATIC PRESSURE IN THE CROSS SECTION OF VEIN.

Now, the question arises: Is there a difference between a given cross section of a stream flowing in a circular conduit and that of a contracted vein flowing from a thin plate orifice into the atmosphere, the flow being horizontal in both instances?

Because of the principle of independence of the effects of forces, in both cases gravity acts on all the elements of each cross section, which are constantly subjected to pressure according to hydrostatic law. In order to facilitate comparison, conditions of flow in a pipe may be so selected that, as shown in Fig. 6, it could be possible to remove a small portion of the pipe without disturbing the stream. In fact, this is considered a classic experiment and is described in text books, such, for instance, in Unwin's Hydromechanics, page 48. The velocity in the pipe and at the contracted section being the same, the jets are similar through the short distance between the parallel planes *ab* and *cd*. Both are surrounded by atmosphere, and it is usually stated in text books that the pressure within the jet is atmospheric. That this is erroneous may be readily proven by means of a finely pointed pitot tube inserted at various points on the vertical diameter of the vein. With four plain tubes inserted as in Fig. 7, the water would rise in each of them to the level *AB*, and the heads would be h_1, h_2, h_3 and h_4. It is therefore quite evident that the actual head causing the flow is about equal to h_1 only, and that the difference

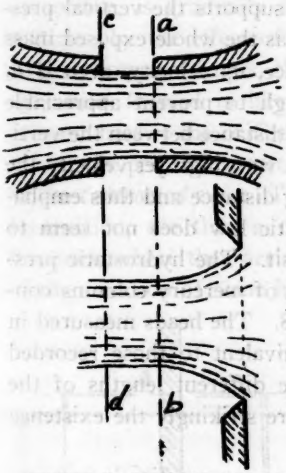


FIG. 6.

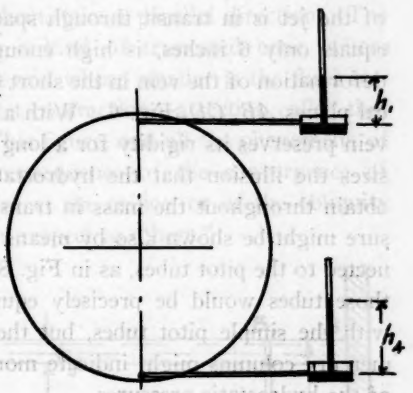


FIG. 8.

(h_4—h_1) represents precisely the height of the column of water standing between the two horizontal planes *CD* and *EF*. It

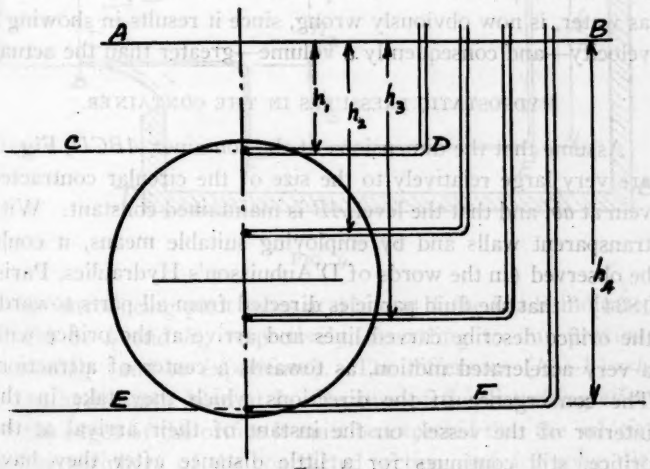


FIG. 7.

follows that the layer in plane EF supports the vertical pressure $(h_4—h_1)$ per unit of area, but as the whole exposed mass of the jet is in transit through space, its velocity, even if h_1 equals only 6 inches, is high enough to prevent appreciable deformation of the vein in the short distance between the vertical planes AB, CD, Fig. 6. With a very high jet velocity the vein preserves its rigidity for a long distance and thus emphasizes the illusion that the hydrostatic law does not seem to obtain throughout the mass in transit. The hydrostatic pressure might be shown also by means of mercury columns connected to the pitot tubes, as in Fig. 8. The heads measured in those tubes would be precisely equivalent to those recorded with the simple pitot tubes, but the different lengths of the mercury columns might indicate more strikingly the existence of the hydrostatic pressures.

It is worthy of remark that the common practice in measuring the flow through a horizontal orifice by means of the pitot tube is to place the point of the tube along the horizontal diameter, the intention being to take into account the *mean* of the hydrostatic pressures. This practice with heavy fluids, such as water, is now obviously wrong, since it results in showing a velocity—and consequently a volume—greater than the actual.

HYDROSTATIC PRESSURES IN THE CONTAINER.

Assume that the dimensions of the container $ABCD$, Fig. 9, are very large relatively to the size of the circular contracted vein at ab, and that the level AB is maintained constant. With transparent walls and by employing suitable means, it could be observed (in the words of D'Aubuisson's Hydraulics, Paris, 1834) "that the fluid particles directed from all parts towards the orifice describe curved lines and arrive at the orifice with a very accelerated motion, as towards a center of attraction. The convergence of the directions which they take in the interior of the vessel, on the instant of their arrival at the orifice, still continues for a little distance after they have passed through it, so that the fluid vein, at its passage from

the orifice, is gradually contracted up to a point where its particles, by the effect of their reciprocal action, and of the motions impressed upon them, take a parallel direction. The vein thus formed is like a kind of truncated cone, whose greatest base is the orifice and whose smaller is the fluid section at the point of greatest contraction—a section called that of the *contracted vein*. This figure, and all the phenomena of contraction, are thus a consequence of the convergence of the lines, where they arrive at the orifice, or of the obliquity of the direction of some in respect to others."

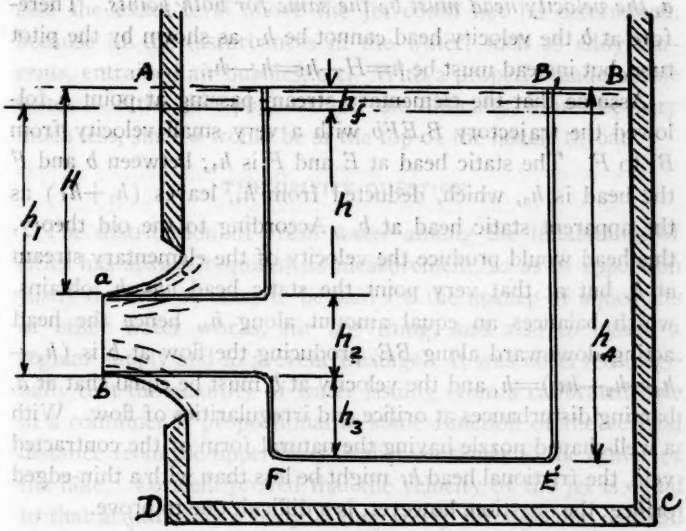


FIG. 9.

By introducing pitot tubes at *a* and at *b* the heads indicated would be h and h_1, respectively, according to our previous reasoning (and to experience). The difference $(h_1—h)=h_2$ is the hydrostatic head at *b*. Because of the disturbing effect of the edges of the orifice it must be expected that the velocity at the periphery of the contracted section, and for some distance radially, is less than the actual velocity expected from

the obtaining flow head. Such a disturbance is obviously responsible for the observed variation of the velocity from the periphery to the axis of the vein. But since, according to Bazin, the lines of equal velocity are practically concentric circles, the points a and b are therefore equally affected by the frictional disturbances of the orifice's edges. Calling h_t the head lost through edge friction and H the vertical distance from a to level AB, it is clear that the pitot head at a must be $h=H-h_t$, and that it is the actual head productive of the velocity at a. Now, *since the same velocity obtains at b as at a, the velocity head must be the same for both points.* Therefore at b the velocity head cannot be h_1, as shown by the pitot tube, but instead must be $h=H-h_t=h_1-h_2$.

Assume that the elementary stream passing at point b followed the trajectory B_1EFb with a very small velocity from B_1 to F. The static head at E and F is h_4; between b and F the head is h_3, which, deducted from h_4, leaves (h_1+h_t) as the apparent static head at b. According to the old theory, this head would produce the velocity of the elementary stream at b, but at that very point the static head $ab=h_2$ obtains, which balances an equal amount along h_4, hence the head acting downward along BE, producing the flow at b is $(h_4-h_3-h_2-h_t)=h$, and the velocity at b must be equal that at a, barring disturbances at orifice and irregularities of flow. With a well-shaped nozzle having the natural form of the contracted vein, the frictional head h_t might be less than with a thin-edged orifice; the question, however, is a difficult one to prove.

SUBMERGED ORIFICE.

Instead of discharging freely into the atmosphere, the jet in the preceding, Fig. 9, might be so arranged as to deliver into another container, the level of water therein being, by suitable means, kept constant at any desirable datum. It seems obvious, then, that until the level reaches the point a, at the top of the most contracted section, the condition of flow is not different in submergence from that in atmosphere, since

the hydrostatic pressure at *ab* will not be affected by the submergence. The case resembles that shown in Fig. 6, where the pipe was opened to the atmosphere.

With the down-stream level at a distance h_a above the point *a*, the flow head would be $(H—h_a—h_t)$. But, this head being smaller than $(H—h_t)$, the contracted section would not have quite the same height *ab* as before, and the form of the jet would be altered; therefore, it would not be possible to determine the exact position of *a*, even experimentally, because the jet would entrain the water surrounding it. And also the exact level above the jet could not be determined, because of the disturbances in the water, such as eddy currents, entrained air bubbles, etc. With a properly made nozzle instead of the thin plate orifice, the difficulty would be very much less, since *a* would be at the top of the nozzle throat.

THE ORIFICE QUESTION.

The distribution of fresh water among the inhabitants of cities has always required its measurement, so as to apportion justly the tolls or taxes to be paid for the upkeep of aqueducts or other water works, for one thing, and also to insure a regular supply and to prevent wastage. It was observed originally that the quantity of water issuing from a calibrated hole in a container is proportional to some function of the vertical distance from the upper level in that container to the center of the hole. Toricelli proved that the velocity of the jet is equal to that acquired by a body falling freely through that vertical distance. Thus, knowing that distance or head, it was easy to compute the amount of flow through an orifice of given area, but the amount thus found was always less than that actually measured. This very point led Newton to the discovery and study of the phenomenon of contraction. Then, also, it was found in Italy that, by providing the orifice with an extension, or nozzle piece, the amount delivered could be made greater than that figured, the increase depending upon, and varying with, the form of the nozzle. All these confusing

questions led mathematicians and experimenters to a tremendous amount of studies and observations, which resulted in establishing for the flow under high heads the laws in present use. But for very low heads and for thin wall orifices, or their equivalents, the matter is still confused. The reasons therefor are that *the real head causing the flow is improperly estimated* and that, per force of circumstances, one of the factors used for determining the volume delivered is *the area of the orifice itself*. If it were possible in all cases to measure the area of the most contracted section, the question would, most probably, have been solved long ago. With a jet issuing at low velocity, it is utterly impossible to locate the exact position of that section, and, furthermore, if found, its surface would be convex instead of plane, and its actual area could not be computed.

The *area of the orifice* is used in the calculations simply because it is a tangible factor and therefore can be accurately measured. But the water does not traverse perpendicularly the plane of the orifice. Thus, as we have seen in Fig. 2, at the edges the tangent to the jet forms an angle of 54 degrees with the plane. It is obvious that this angle goes on diminishing from *M* towards *o*, Fig. 10, on the horizontal axis, where it becomes zero; thence it increases, until at *N* it is again 54 degrees, for this particular instance. A curve *MoN*, extending from the upper to the lower edges of the orifice may be considered as the generatrix of a convex surface, which all the elementary streams flowing towards *ab* cross perpendicularly. The area of this surface is obviously much greater than that of the orifice itself. Were it not for the retarding effect of friction of the water against the edges, the streams would all have the same velocity when traversing that surface; for, taking the conditions at the two extreme points, *M* and *N*, we must have, according to the Bernouilli theorem, at *M* the *static head h plus the flow head* h^1, *and at N, the static head h plus the static head MN plus the flow head* h^1. The same reasoning would apply to the comparison of points *M* and *o*, *o* and *N*.

ANNULAR ORIFICE.

A case most worthy of remark is that of annular jet which is found very readily when the plane of the orifice is horizontal, the head being usually very small. The phenomenon occurs to a lesser degree with a lateral orifice. The upper level is seen at first to assume a dish shape, and then the center is quickly depressed until a funnel is formed, which extends from the level to the *center* of the orifice itself. When the orifice is continued by a pipe a loud gurgling noise is heard.

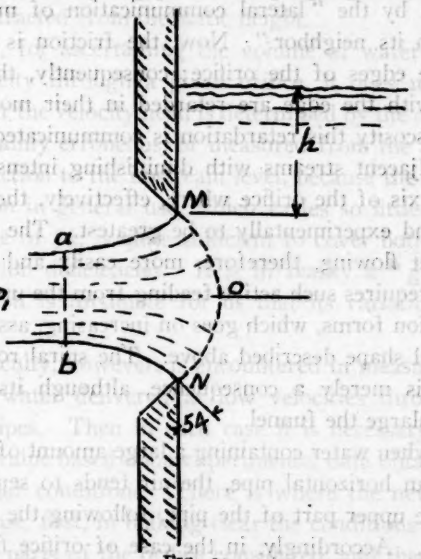


FIG. 10.

The phenomenon is complicated in that the liquid "walls" of the funnel are set in rapid rotation, thus setting up intense centrifugal actions, which accounts for the relatively large size of the funnel. The rotation is either right or left-handed; it may be stopped by means of thin planes held vertically along the funnel path. It is obvious that the air filling the funnel is entrained by contact with the "walls" and that it passes through the orifice along with the hollow jet.

The explanation of the phenomenon is very easy, but the

writer is not aware that it has been presented before in such a simple form. *It takes place because the fluid is imperfect;* that is, because it is viscous and is therefore subject to friction within itself; and, finally, because the walls of the orifice and of the container are, however smooth, themselves imperfect and subject to friction. The sagacious Bossut remarks that "the roughness of the walls of the container contributes also to increase more or less the funnel." Venturi, in 1787, developing the theoretical ideas of Newton, showed that all is explained by the "lateral communication of motion of one stream to its neighbor." Now, the friction is most intense along the edges of the orifice; consequently, the streams in contact with the edge are retarded in their motion; because of the viscosity this retardation is communicated successively to the adjacent streams with diminishing intensity from the edge to axis of the orifice where, effectively, the velocity has been found experimentally to be greatest. The axial portion of the jet flowing, therefore, more easily and rapidly than the rest, requires such active feeding from the upper level that a depression forms, which goes on increasing, assuming finally the funnel shape described above. The spiral rotation of the "walls" is merely a consequence, although its effect is to greatly enlarge the funnel.

Now, when water containing a large amount of air is forced through an horizontal pipe, the air tends to separate and to rise to the upper part of the pipe, following the path of least resistance. Accordingly, in the case of orifice flow it would seem that the funnel should lead the air along Ma, whereas it always leads it along the horizontal axis oo_1, which is additional proof that the velocities throughout the jet issuing from the orifice are adjusted along concentric circles—or figures—of equal intensities, as observed by Bazin.

CONCLUSIONS.

This analysis, extended, perhaps, to the point of tediousness, but for the purpose of making the demonstration as complete as possible, enables us to put forth the following

theorem: *The velocity of flow of water issuing from a container through a lateral orifice into atmosphere is generated by the head, or static height, measured from the top of the contracted section of the issuing vein to the upstream level in the container.* This supposes, of course, that the pressure upon the upstream level is that of the atmosphere, but it frequently occurs that that pressure is greater or less than that of the medium surrounding the issuing vein, and then the difference of pressure, expressed in terms of head, must be added or subtracted from the static height.

Ordinarily, for ascertaining the volume of water delivered at high velocity through a given orifice, it does not matter much whether the velocity head is determined by the old theory, which is certainly erroneous, or measured from the top of the contracted section to the upstream level, because the coëfficient of contraction in general use, which varies so little from the average value of $\frac{5}{8}$, is quite sufficient to cover both the head and contraction deficiencies. It is in reality a "global" coefficient, and it is fortunate for us that its variations are so restricted.

Much difficulty, however, is encountered in measuring large volumes of water delivered at low velocities through large orifices or pipes. Then in each case it is necessary to select a coëfficient value based upon experimental data obtained from nearest similar conditions. There is where the new theorem becomes of use, first, in making clear the conditions of flow in the different parts of the orifice considered, and then in facilitating the establishment of new and more rational coëfficients.

This theorem has proven of great value in laying out piping plans for pumping installations, condensers, irrigation apparatus, etc. These may be termed "discontinuous systems," because the flow must be considered as taking place independently from one part to another, from that one to the following one, etc.; and where sizes of fittings, valves and connections, differing from those of the piping, preclude uniformity of velocity of flow throughout the system.

TEST OF REILLY MULTICOIL EVAPORATOR (SUBMERGED TYPE).

By H. T. Dyer, Lieutenant Commander, U. S. Navy, Member.

A portion of the results of a test on a Reilly multicoil evaporator (submerged type) made during 1916 and covering a period of about ten months is published to afford a comparison between this type and a later test of a different type, the Schutte and Koerting evaporator, reported in the November, 1917, number of the Journal of the American Society of Naval Engineers, and also to bring out certain points of interest, and it is hoped, of value, to those having the operation of the Reilly multicoil evaporator under their supervision, there being large numbers of these evaporators installed in our latest ships. Although this particular evaporator is not in all details exactly the same as the present type, the results obtained and general conclusions are, however, applicable to the type of evaporator now being installed.

The general arrangement of the evaporator is shown in Fig. 1, and indicates a cast-iron shell as shown. The heating elements consist of the usual "Reilly" coils connected to two horizontal headers at the top and two at the bottom, the coils in this particular evaporator being considerably shortened so as to operate with them completely submerged, hence the trade name "Submerged Type."

Attention is invited to the rigid casting forming the connection between the top and bottom headers at the rear (Fig. 2), this rigid connection being provided so that the expansion of the coils due to changes in temperature will not separate the headers, but will be taken up in the coils themselves and so break off the scale that may be formed.

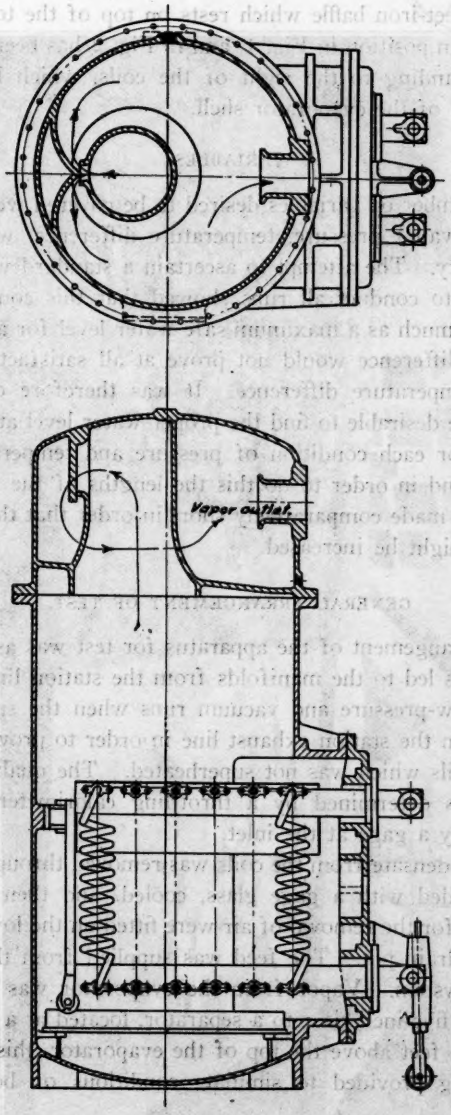


FIG. 1.

The sheet-iron baffle which rests on top of the top headers is shown in position in Fig. 1, and in Fig. 2 has been removed and is standing to the right of the coils, which have been pulled out of the evaporator shell.

VARIABLES.

The number of variables desired to be covered were: steam pressure, vapor pressure, temperature difference, water level and salinity. The attempt to ascertain a standard water level at which to conduct all runs showed that this could not be done, inasmuch as a maximum safe water level for a low temperature difference would not prove at all satisfactory for a higher temperature difference. It was therefore considered to be quite desirable to find the proper water level at which to operate for each condition of pressure and temperature difference, and in order to do this the lengths of the individual runs were made comparatively short in order that the number of runs might be increased.

GENERAL ARRANGEMENT OF TEST.

The arrangement of the apparatus for test was as follows: Steam was led to the manifolds from the station lines except for the low-pressure and vacuum runs when the supply was taken from the station exhaust line in order to provide steam for the coils which was not superheated. The quality of the steam was determined by a throttling calorimeter and the pressure by a gage at the inlet.

The condensate from the coils was removed through a drain pot, provided with a gage glass, cooled, and then weighed. Air cocks for the removal of air were fitted on the lower manifold and drain pot. The feed was supplied from the station flushing system. Vapor from the evaporator was taken off through a five-inch riser to a separator, located at a height of about nine feet above the top of the evaporator, this arrangement being provided to simulate conditions on board ship

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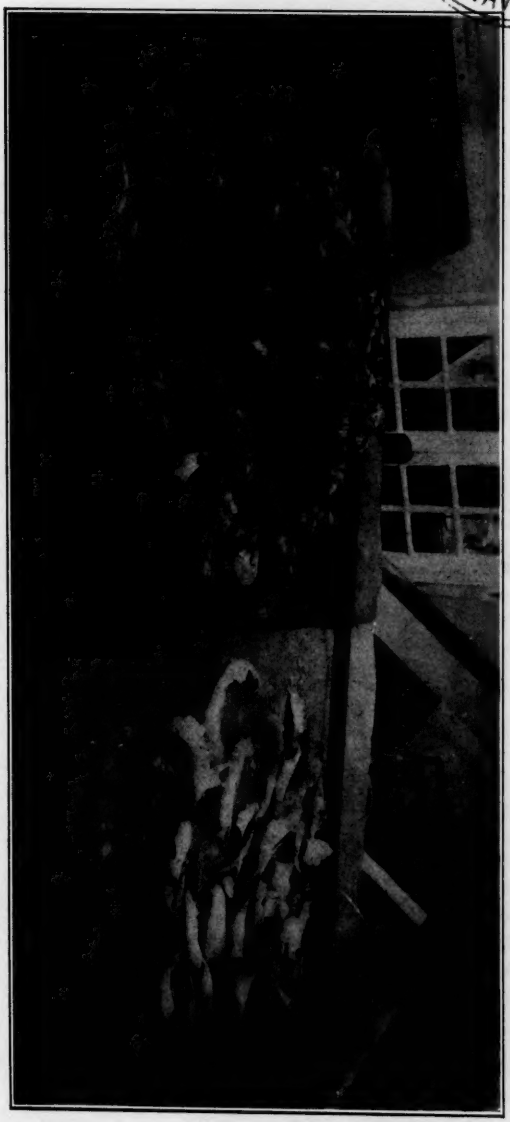


FIG. 13.

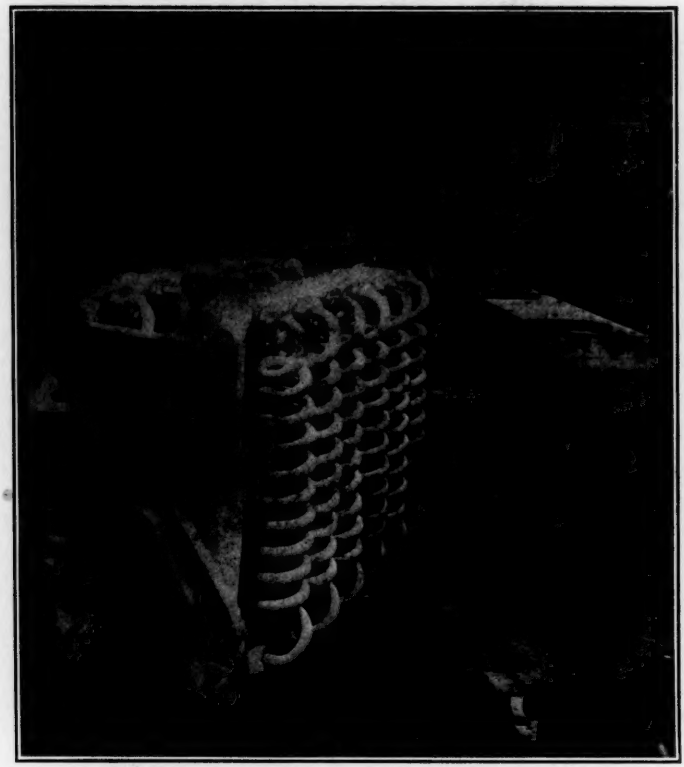


FIG. 2.

where the distiller is placed at a considerable height above the evaporator. From the separator the vapor was led to a surface condenser, condensed, and then weighed.

The vapor pressures in the shell and dome were measured by means of mercury columns and the temperature in the dome was measured by means of a thermometer in a well in the top of the dome. The foregoing description applies to runs in which the coil pressure was above five pounds. For the runs in which the coil pressures were atmospheric and below the evaporator was moved to another location where the drain-pot discharge was connected to a separate condenser in order to provide the necessary means of producing low pressures on the coils.

The procedure during any day included cracking the scale at the beginning and end of the day but not between runs. On starting in the morning the steam line was first drained of condensate and then about sixty or seventy pounds pressure was quickly built up on the coils, while the shell was full of cold water above the top header. The vapor valve being closed, a pressure of fifteen to twenty pounds was built up on the shell, when the blow-down valve was opened, blowing out brine and loose scale. At the end of the day this operation was repeated from one to three times, depending on the capacity developed during the day. When exhaust steam was used about six or seven pounds was the greatest available pressure, and the scale was cracked by turning full steam on the coils and then pumping the shell full of cold water as fast as possible, causing a chilling of the coils. While this "cooling method" was probably the best when only low steam pressures were available it was undoubtedly not as effective as the first mentioned or "heating method" because of the slow creeping of the water up around the coils. During the "heating method" a vibration lasting four or five minutes accompanied the sudden change in temperatures and probably made the cracking of the scale still more effective.

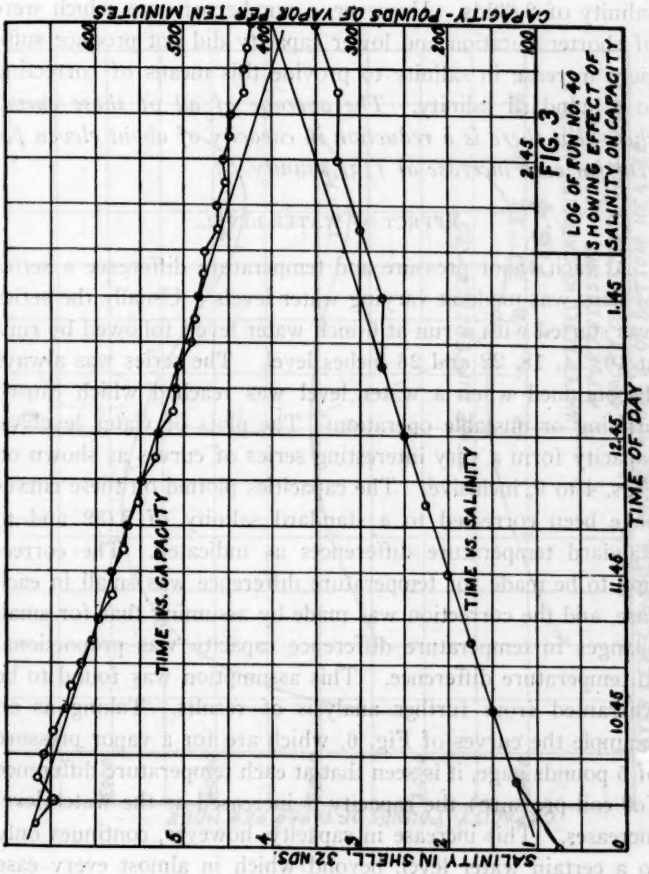
It was desired to start each run with a shell salinity of 1/32, and to blow down when the salinity reached 3/32, thus maintaining an average salinity of 2/32. Inasmuch as feed water with a salinity of 1/32 was not available, each run was started by filling the shell to the desired water level with a mixture of brine, saved from a previous blow down, and Severn River water, which has a salinity of from ⅛ to ¼ of one thirty-second. It was impossible to adjust the salinity at the start of the run to exactly 1/32 by this method, but the salinity was brought as near 1/32 as possible. For certain runs of low capacity the salinity at the start of a run was not brought to 1/32 but was kept at a higher value, in order that the average salinity for the run would be near 2/32. During the run Severn River water was used as feed.

During the runs having vapor pressures above the atmosphere the salinity was taken every half hour by readings of the standard salinometer. For runs in which the shell pressure was below the atmosphere samples were drawn from the shell into a closed bottle and the salinity found by density determinations with the Westphal balance.

Observations were taken every ten minutes for one and one-half hours except in a few cases where runs were curtailed or extended because a slight initial priming left a sediment of salt in the vapor pipe and condenser. A few runs imposing exceptionally severe conditions were discontinued because of violent priming.

EFFECT OF SALINITY IN SHELL.

It has always been recognized that, all other operating conditions being constant, the capacity of an evaporator is affected considerably by the salinity of the water in the shell. In order to determine the effect of salinity on capacity and in order that all results might be corrected to a standard average salinity of 2/32 several runs were made extending over the entire day. The log of one such run is shown on Fig. 3. These logs show steadily increasing salinity and decreasing


CAPACITY - POUNDS OF VAPOR PER TEN MINUTES
600
500
400
300
200
100
TIME VS. CAPACITY
TIME VS. SALINITY
SALINITY IN SHELL, 32 NDS.
6
5
4
3
2
1
0
2.45
1.45
12.45
11.45
10.45
TIME OF DAY
FIG. 3
LOG OF RUN NO. 40
SHOWING EFFECT OF
SALINITY ON CAPACITY

capacity. In addition to these six-hour runs the log of each separate run was plotted similarly in order to determine the relation between salinity and capacity. This provided an automatic means of correcting the capacity to the standard salinity of 2/32ds. However, a number of runs which were of shorter duration and lower capacity did not produce sufficient increase in salinity to provide this means of correcting to a standard salinity. *The average of all of these curves show that there is a reduction in capacity of about eleven per cent for each increase of 1/32 salinity.*

EFFECT OF WATER LEVEL.

At each vapor pressure and temperature difference a series of runs was made at varying water levels. Usually the series was started with a run at 6-inch water level, followed by runs at 10, 14, 18, 22 and 26 inches level. The series was always discontinued when a water level was reached which caused priming or unstable operation. The plots of water level vs. capacity form a very interesting series of curves as shown on Figs. 4 to 9, inclusive. The capacities plotted on these curves have been corrected to a standard salinity of 2/32 and to standard temperature differences as indicated. The correction to be made for temperature difference was small in each case, and the correction was made by assuming that for small changes in temperature difference capacity was proportional to temperature difference. This assumption was found to be warranted from further analysis of results. Taking as an example the curves of Fig. 6, which are for a vapor pressure of 5 pounds gage, it is seen that at each temperature difference (or coil pressure) the capacity is increased as the water level increases. This increase in capacity, however, continues only to a certain water level, beyond which in almost every case there is a small decrease in capacity as the water level is increased. A further increase in the water level results in sudden priming of the evaporator and unstable conditions.

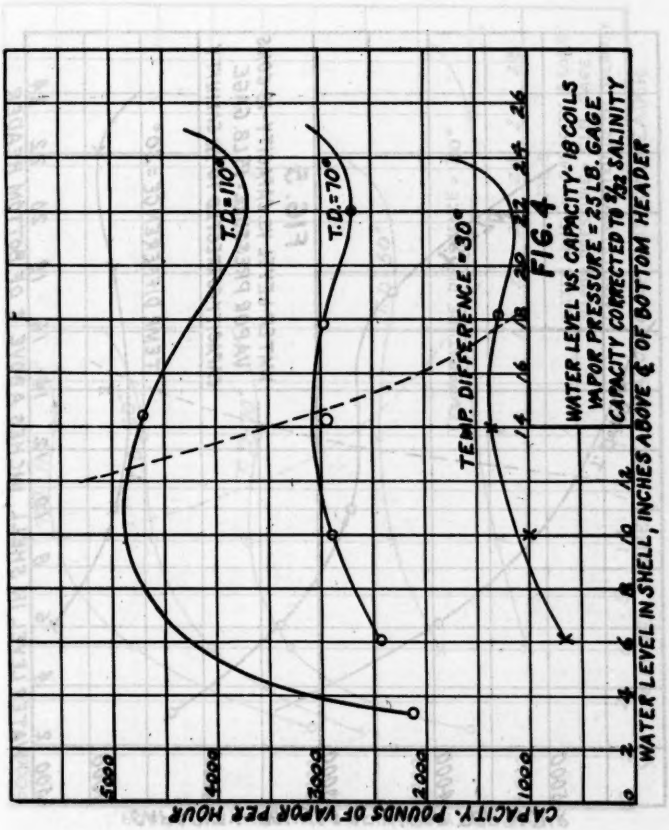


FIG. 4

WATER LEVEL VS. CAPACITY - 18 COILS
VAPOR PRESSURE = 2.5 LB. GAGE
CAPACITY CORRECTED TO 3/32 SALINITY

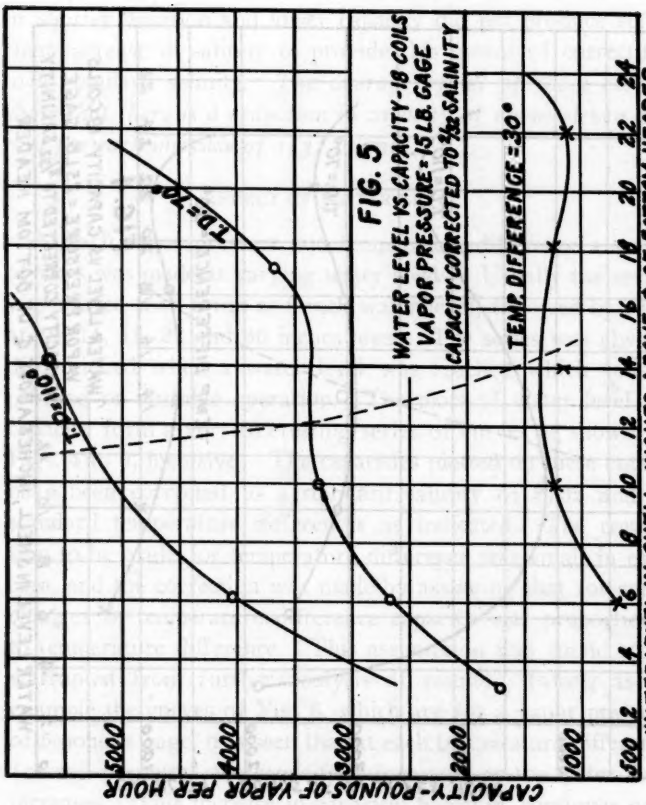


FIG. 5
WATER LEVEL VS. CAPACITY-18 COILS
VAPOR PRESSURE = 15 LB. GAGE
CAPACITY CORRECTED TO 3% SALINITY
TEMP. DIFFERENCE = 20°
T.D. = 20°
T.D. = 10°
CAPACITY - POUNDS OF VAPOR PER HOUR
5000
4000
3000
2000
1000
500
2
4
6
8
10
12
14
16
18
20
22
24
WATER LEVEL IN SHELL, INCHES ABOVE ℄ OF BOTTOM HEADER

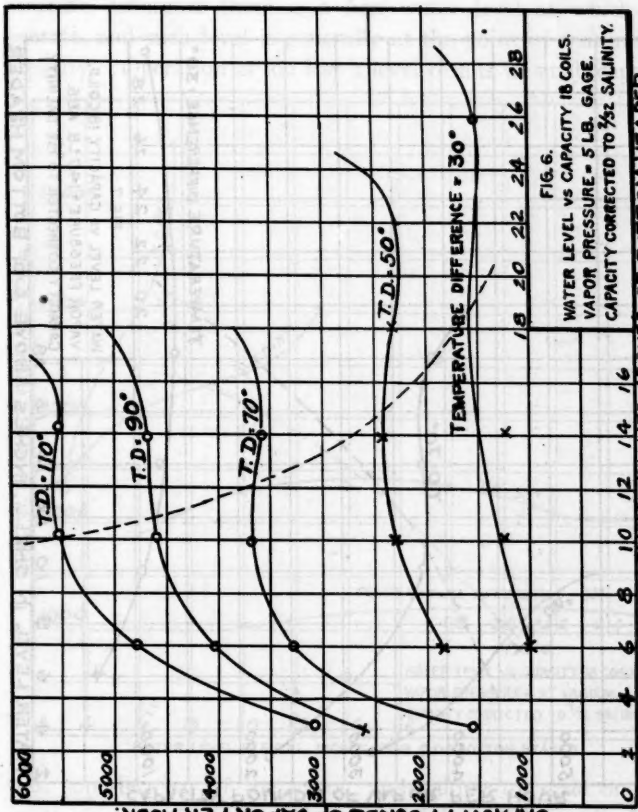
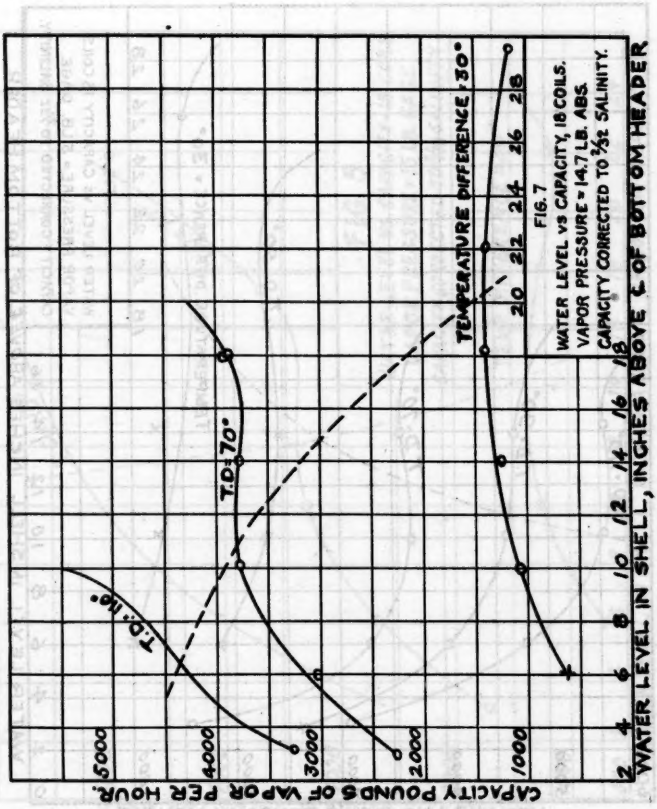
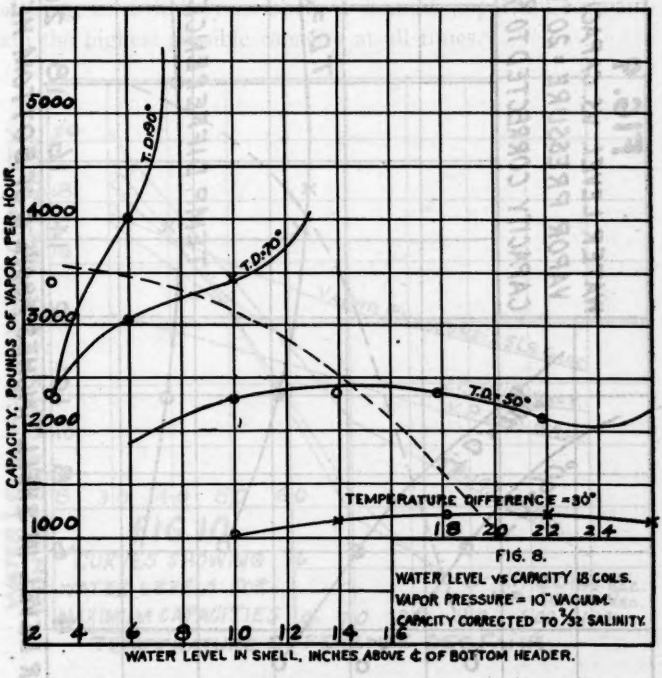


FIG. 6.
WATER LEVEL vs CAPACITY 18 COILS.
VAPOR PRESSURE = 5 LB. GAGE.
CAPACITY CORRECTED TO 3/32 SALINITY.


CAPACITY POUNDS OF VAPOR PER HOUR.
5000
4000
3000
2000
1000
T.D.=10°
T.D.=70°
TEMPERATURE DIFFERENCE, 30°
2 4 6 8 10 12 14 16 18 20 22 24 26 28
WATER LEVEL IN SHELL, INCHES ABOVE ℄ OF BOTTOM HEADER
FIG. 7
WATER LEVEL vs CAPACITY, 18 COILS.
VAPOR PRESSURE = 14.7 LB. ABS.
CAPACITY CORRECTED TO 3½ SALINITY.

It was also observed that the chlorine content of the water was very low, *generally below 0.5 of a grain per gallon and constant over the whole range of water level until the priming level was reached.* These results indicate that at each temperature difference there is a *best* water level at which to operate, and such level is generally at the point of maximum capacity. Operation at too low a level results in small capaci-



ties, superheating of the vapor, and baking of scale on the upper part of the coils. Operating at levels above the best level does not increase the capacity and endangers priming. On Figs. 4 to 9, inclusive, have been drawn dotted lines showing the probable best water levels at which to operate. By operating at water levels as indicated by the dotted lines the maximum capacity is obtained, the vapor is of excellent quality

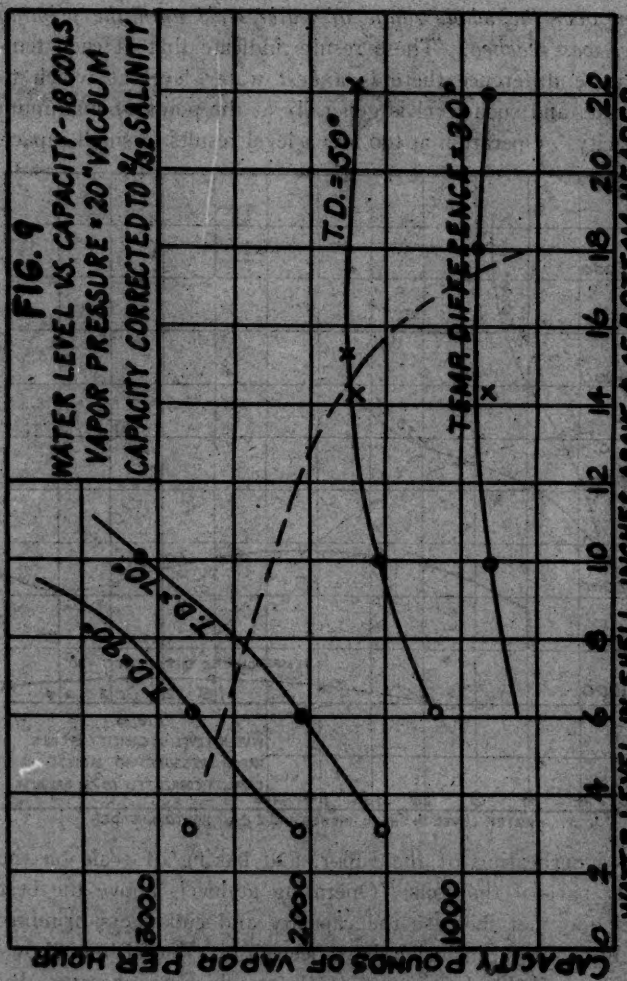


FIG. 9
WATER LEVEL VS. CAPACITY - 18 COILS
VAPOR PRESSURE = 20" VACUUM
CAPACITY CORRECTED TO 3/32 SALINITY
T.D. = 50°
TEMP. DIFFERENCE = 30°
0.1" 0.2" 0.3" 0.4"
CAPACITY POUNDS OF VAPOR PER HOUR
WATER LEVEL IN SHELL, INCHES ABOVE ℄ OF BOTTOM HEADER
0
1000
2000
3000
2
4
6
8
10
12
14
16
18
20
22

and the water level is sufficiently far removed from the priming point to remove the danger of sudden priming.

Attention is particularly invited to the wide range of water levels that may be employed with very small changes in capacity under almost all conditions and no change in chlorine content of the distillate, a very desirable feature in an evaporator installed on a small vessel subjected to violent rolling and pitching, as a destroyer, where it is most important to maintain the highest possible capacity at all times.

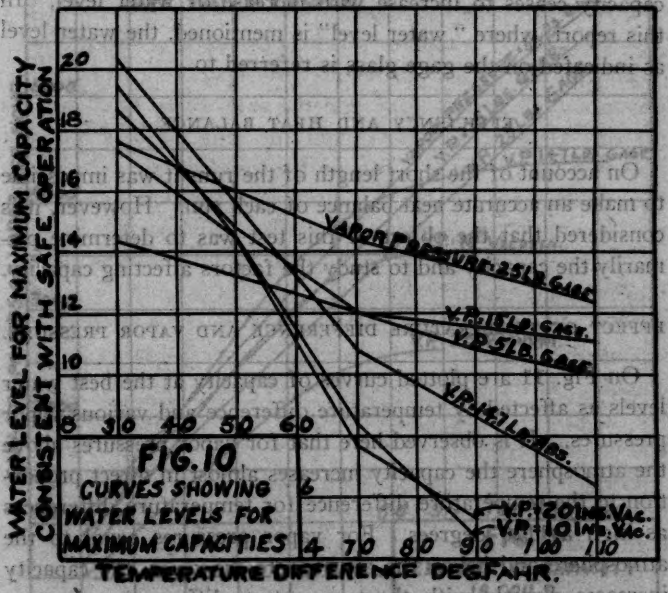


FIG. 10
CURVES SHOWING WATER LEVELS FOR MAXIMUM CAPACITIES

These best water levels have been summarized on Fig. 10, which figure shows the relation between vapor pressures, temperture differences and water levels.

Attention is invited to the fact that the water levels here recorded are the levels as indicated in the gage glass on the shell of the evaporator, and that the vertical distance between centers of top and bottom steam manifolds was 24 inches. It

is evident that the actual water level in the shell is not the same as the level indicated on the gage glass outside of the shell. On account of the boiling taking place in the shell, the water in the shell is considerably less dense than the water in the gage glass outside, and the level indicated by the gage glass is lower than the actual level in the shell, and there is probably an average difference between the two levels for various conditions, of 10 to 12 inches. It is altogether probable that the coils are completely submerged at that point at which the capacity ceases to increase with increase of water level. In this report, where "water level" is mentioned, the water level as indicated on the gage glass is referred to.

EFFICIENCY AND HEAT BALANCE.

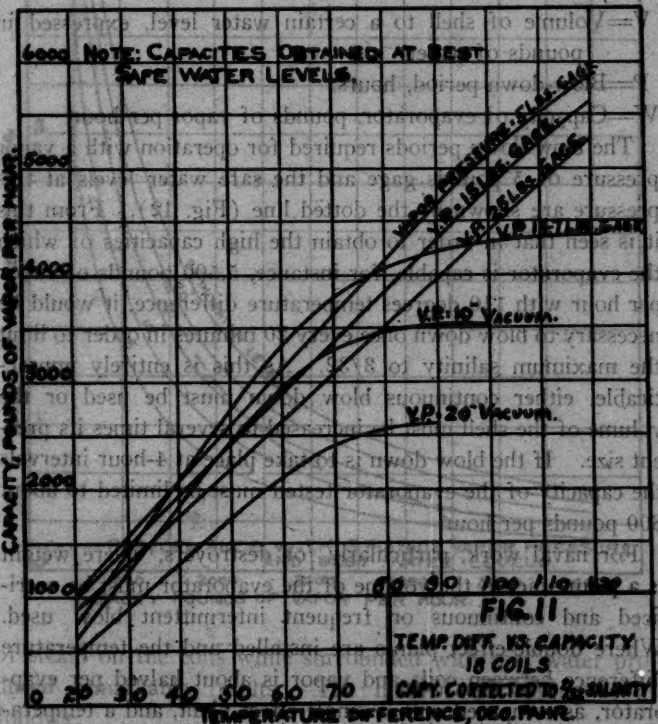
On account of the short length of the runs it was impossible to make an accurate heat balance of each run. However, it is considered that the object of this test was to determine primarily the capacity and to study the factors affecting capacity.

EFFECT OF TEMPERATURE DIFFERENCE AND VAPOR PRESSURE.

On Fig. 11 are plotted curves of capacity at the best water levels as affected by temperature difference and various vapor pressures. It is observed here that for vapor pressures above the atmosphere the capacity increases almost in direct proportion to the temperature difference for temperature differences as high as 120 degrees. For vapor pressures equal to the atmosphere and below the atmosphere, however, the capacity increases directly with the temperature difference only to a temperature difference of 60 or 70 degrees, beyond which there is very little increase in capacity with increasing temperature difference. This is probably due to the fact that an excessive amount of surface for the low vapor pressures necessitates the carrying of very low water levels in order to prevent priming with high temperature differences.

BLOW-DOWN PERIOD.

The requirements as to blow-down period are as follows: "The design of evaporators should be such that there is sufficient water space in the evaporator to maintain saturation below 3/32 without requiring blowing down more often than about once in four hours. . . ." The blow-down periods as



determined on this test are not applicable to operation with sea water, inasmuch as the salinity of Severn River water is from ⅛ to ¼ that of sea water, and in a given period of time only ⅛ to ¼ as much salt would be fed into the evaporator as with sea water feed. The blow-down periods which would be

required when operating with sea water and a maximum salinity of 3/32 have been computed and are plotted on Fig. 12. The blow-down periods were computed from the formula:

$$\frac{V}{32}+\frac{WP}{32}=\frac{3}{32}V;\ \text{or}\ P=\frac{2V}{W}.$$

in which

V=Volume of shell to a certain water level, expressed in pounds of water.

P=Blow-down period, hours.

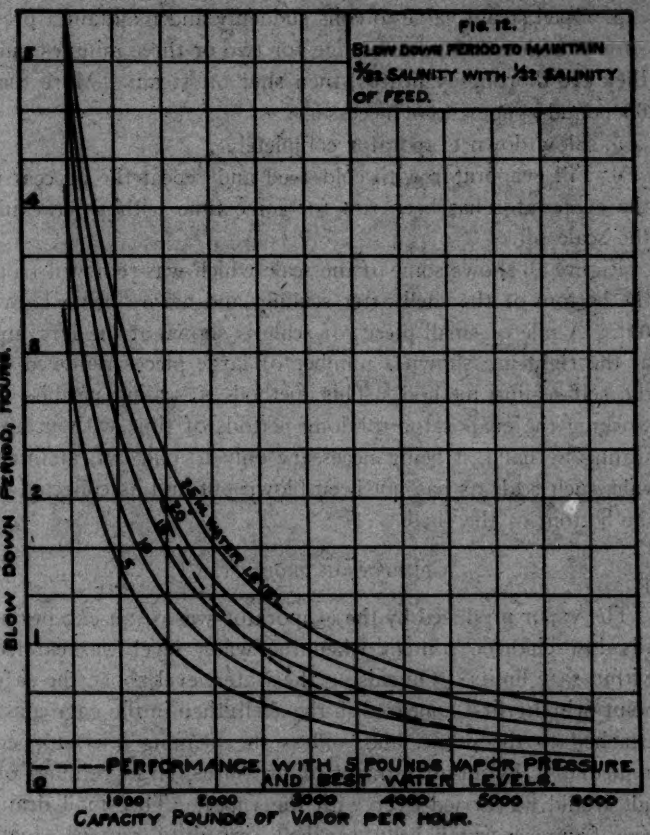
W=Capacity of evaporator, pounds of vapor per hour.

The blow-down periods required for operation with a vapor pressure of 5 pounds gage and the safe water levels at this pressure are shown by the dotted line (Fig. 12). From this it is seen that in order to obtain the high capacities of which the evaporator is capable, for instance, 5,500 pounds of vapor per hour with 110 degrees temperature difference, it would be necessary to blow down once every 20 minutes in order to limit the maximum salinity to 3/32. As this is entirely impracticable, either continuous blow down must be used or the volume of the shell must be increased to several times its present size. If the blow down is to take place at 4-hour intervals the capacity of the evaporator tested must be limited to about 800 pounds per hour.

For naval work, particularly for destroyers, where weight is a prime factor, the volume of the evaporator must be sacrificed and continuous or frequent intermittent blow used. Where double-effect plants are installed and the temperature difference between coils and vapor is about halved per evaporator, as compared with a single-effect plant, and a temperature difference of say 45 degrees F. used, the blow-down period is materially increased.

SELF SCALING OF COILS.

The method of cracking the scale by the sudden application



of steam on the coils while surrounded with cold water produced remarkable results. The method of scaling is as follows:

1. Close steam valve to coils and blow down evaporator completely.

2. Close bottom blow valve.

3. Fill evaporator with *cold* feed until coils and top headers are completely submerged.

4. Open steam valve to coils suddenly and maintain a pressure of at least 40 pounds gage for two or three minutes until they are thoroughly heated, then shut off steam. More than 60 pounds gage is not necessary.

5. Blow down evaporator completely.

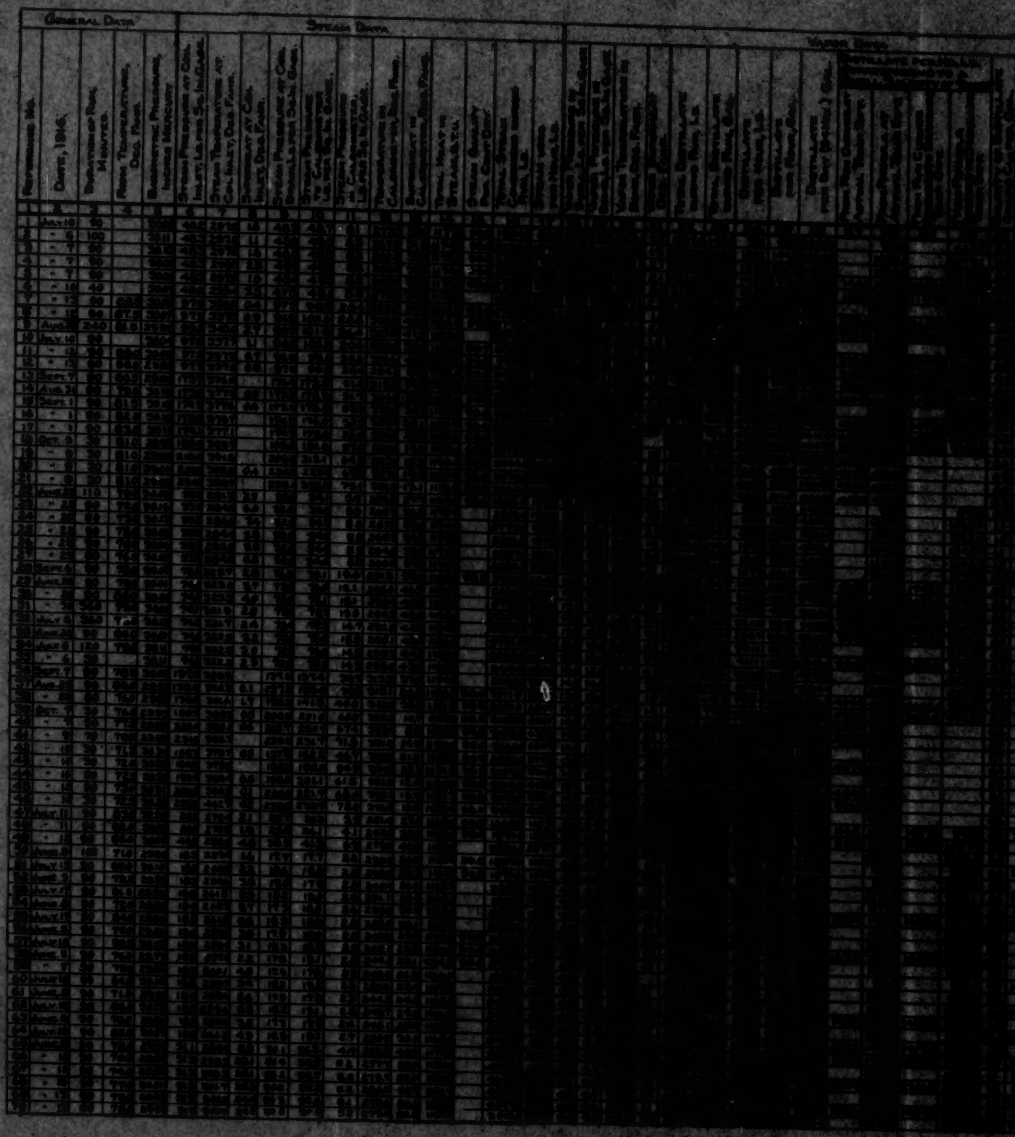
6. Fill evaporator with cold feed and repeat the process if the evaporator has been run for some time without cracking the scale off.

Figure 13 shows some of the scale which was removed from the bottom of the shell after scaling, and before being blown out. A pile of small pieces of scale is shown at the left, and at the right are shown a number of large pieces removed by the self-scaling method. This method of scaling permits of working the evaporator for long periods of time without any scaling by hand, it being necessary only to pull out, at intervals, such scale as has not been blown out and as collected in the bottom of the shell.

PURITY OF PRODUCT.

The vapor produced by the evaporator was of an exceptionally low chlorine content when the water level was carried within safe limits. On raising the water level above the safe point priming first appeared in the shell, then in the gage glass attached to the dome, and, unless the priming was stopped immediately by an adjustment of pressures, large volumes of salt would be carried over with the vapor. The small drain pipe from separator head to shell served to remove small quantities of salt when the priming was not violent. The appearance of water in the gage glass on the dome would frequently enable an adjustment of conditions to be made which would prevent salt being carried over to the condenser. The dome does not, therefore, prevent the carrying over of salt, but serves as a partial safeguard.

The principal results and deductions from the test are as follows:



(*a*) There is an average reduction in capacity of about eleven per cent for each 1/32 increase of salinity.

(*b*) For each number of coils, vapor pressure and temperature difference there is a best water level for maximum capacity and safe operation with low chlorine content and no danger of priming.

(*c*) For vapor pressures below the atmosphere the number of coils must be decreased to insure satisfactory operation.

(*d*) In order to obtain a capacity of 5,500 pounds per hour with this evaporator the blow down must occur at intervals of 20 minutes. To avoid this too frequent blow down either continuous blow down must be used or the water space of the evaporator must be greatly increased.

(*e*) The construction of the coils is such as to provide a very effective means of self scaling of coils by sudden changes in temperature.

(*f*) The vapor produced is of exceptionally low chlorine content when the water level and temperature difference is within safe limits.

APPENDIX I.

Figure 14 gives the log of a portion of the runs from which the preceding material was derived. The source of certain items shown on this figure which may not be clear is explained below.

Item 23. Total distillate during run, gallons, $= \frac{\text{Item 22}}{8.328}$.

Item 28. Distillate per hour, pounds, corresponding to a shell salinity of $\frac{1}{32}$ds computed as follows:

$$\text{Item 28} = \text{Item 24} - [\text{Item 24} \times .113\ (\tfrac{1}{32} - \text{Item 42})].$$

Item 29. Distillate per hour, pounds, corresponding to $\frac{1}{32}$ds shell salinity and to desired temperature difference.

$$\text{Item 29} = \frac{\text{Item 27} \times \text{Item 36}}{\text{Item 35}}.$$

This assumes that for small variations in T.D. capacity is directly proportional to temperature difference.

Item 30. Distillate per hour, pounds, corresponding to $\frac{1}{32}$ds shell salinity (by 11.3 per cent correction method) and to desired temperature difference.

$$\text{Item 30} = \frac{\text{Item 28} \times \text{Item 36}}{\text{Item 35}}.$$

Item 38. Theoretical efficiency.

$$e = \frac{H_s - Q_d}{H_v - Q_f + \frac{f}{m - f}\left(t_v - t_f\right)}.$$

where e = theoretical efficiency, pounds of vapor per pound of steam;
H_s = total heat in steam;
Q_d = heat of liquid in coil drain;
H_v = total heat in vapor;
Q_f = heat of liquid in feed water;
f = salinity of feed;
m = maximum salinity obtained;
t_v = temperature of vapor;
t_f = temperature of feed water.

CONDENSER TUBE FAILURE—PHOTOMICROGRAPHY.

By F. W. Sterling, Lieut. Commander, U. S. N., Member.

The usefulness of the photomicrograph in diagnosing the cause of condenser tube leaks was recently clearly demonstrated by a series of microphotographs taken of two successive failures of condenser tubes on the same vessel. The work was done by Professor William Campbell, of Columbia University, temporarily employed as metallurgist at the Navy Yard, New York.

CASE A.

In the first case, three pieces of tube were received, 5/8 inch in diameter, 1/16 inch thick. The composition, admiralty metal, of the tubes was, Cu 70 per cent, Zn 29 per cent, Sn 1 per cent. The first tube showed a seam about 2 inches long, with a hole near one end, shaped like an arrow head, Fig. 1. The second showed a bad seam or lap about 5 inches long, Fig. 2. The third showed three flaws on one side, in line, and one flaw on the other. This third tube was not used in the test.

Figs. 3, 4 and 5 are cross sections cut through the first tube at *a-a*, *b-b* and *c-c*, respectively. Figs. 6 and 7 are two 1-inch lengths cut from the two tubes at *d* and *e* respectively. Fig. 6 was compressed to 5/8 inch and polished as shown. The crack opened up badly on compression. Fig. 7 was compressed to 7/16 inch without cracking.

There had been numerous condenser tube failures on this vessel and the commanding officer had put forward the theory that they were season cracks pure and simple, that had occurred in cold-drawn metal. It is probable that the navy yard authorities concurred in this opinion, in fact it would have been the natural conclusion in the premises.

Fig. 6 Flattened to 3/4"

Fig. 7 Flattened to 1/2"

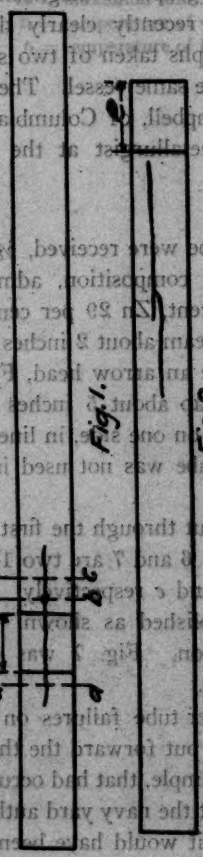


Fig. 1.
Fig. 2
Fig. 3 Section at a-a

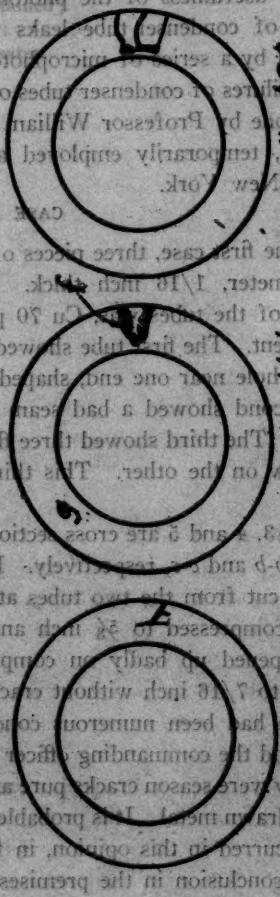


Fig. 5 Section at b-b
Fig. 4 Section at c-c

FIG. 8.

FIG. 9.

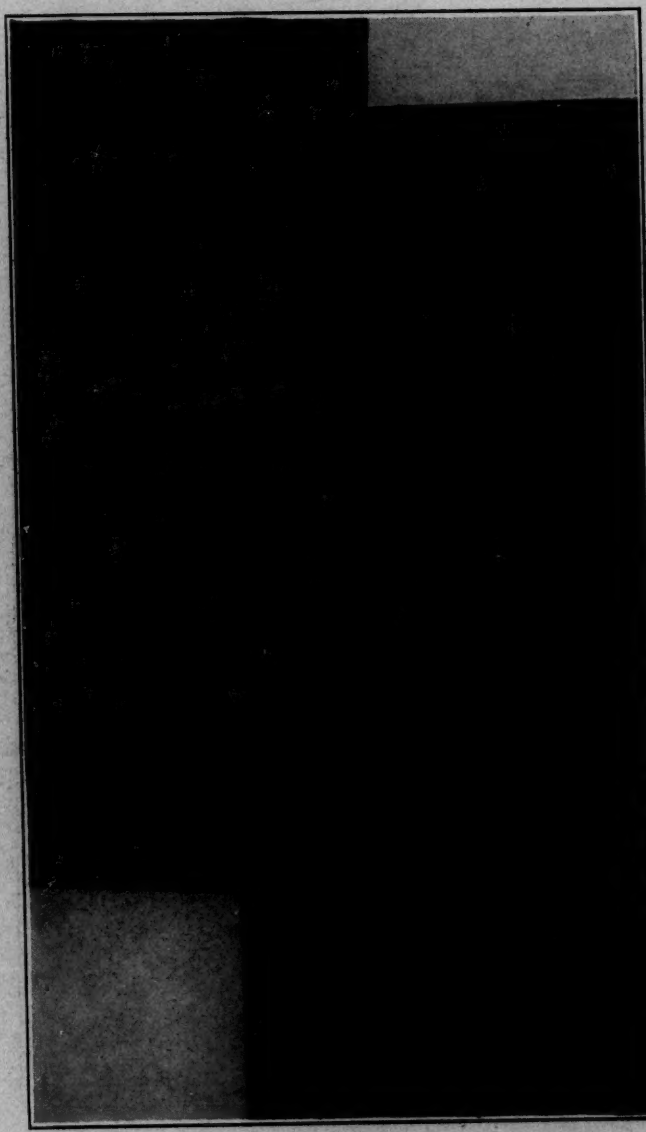


FIG. 10.

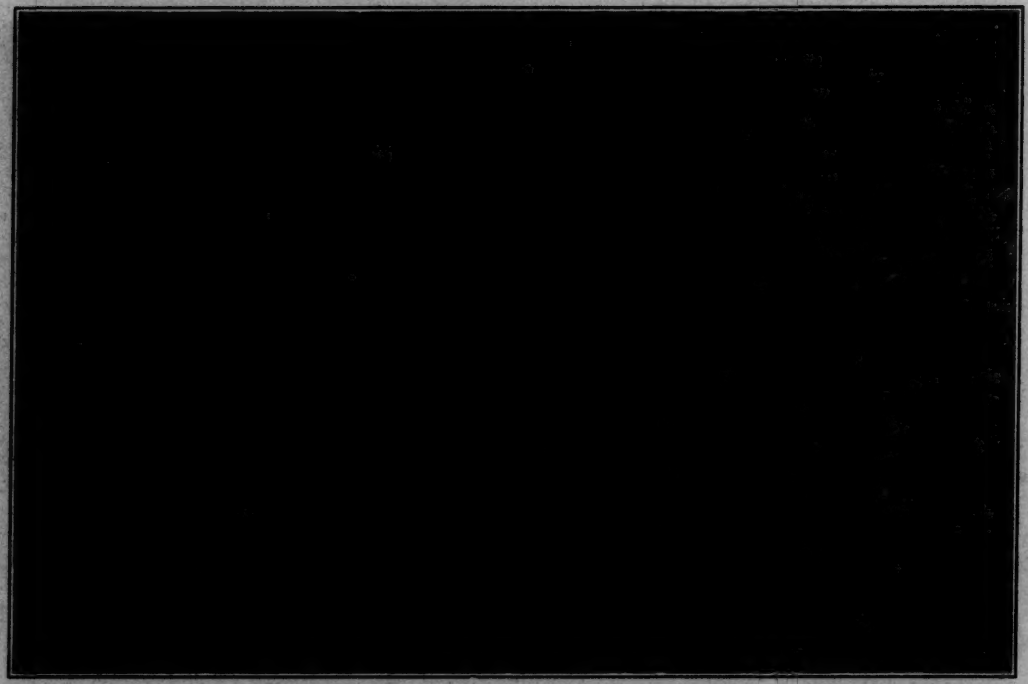


FIG. 11.

FIG. 12.

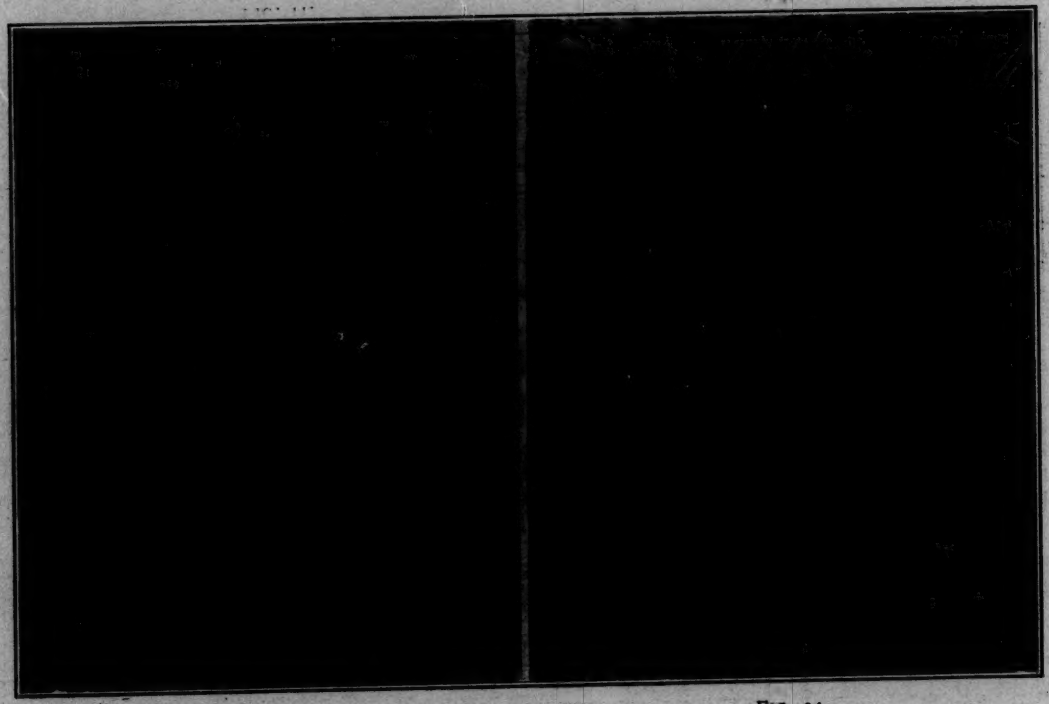


FIG. 13.

FIG. 14.

Fig. 8 shows a branch fissure, as shown in Fig. 3. The microphotograph, magnified 90 diameters, has been repolished to remove etching stains. The structure is that of properly annealed brass.

Fig. 9 shows a V-shaped hole, as shown in Fig. 4. This microphotograph, magnified 90 diameters, was taken on the edge of the hole at *f*, the dark portion of the figure being the hole. The metal is quite normal. A similar photomicrograph taken at *g*, showed a similar structure of metal.

Fig. 10 shows the three fissures, as shown in Fig. 5, all ending in the corroded spot shown in the left side of Fig. 10. This figure is magnified 90 diameters and repolished to remove etching stains. The tube edge appears at the two right-hand corners. Corrosion is present at the base of the cracks. The metal shows a normal structure.

Fig. 11 (×90 diameters) shows the structure of the metal at the crack *k*, Fig. 6; the crack opened out in flattening the test piece, the grains on each side of it being perfectly normal.

Fig. 12 (×90 diameters) taken at *m*, Fig. 6, near the outside, shows a number of fine seams in normal metal.

This figure and another microphotograph taken at *p*, Fig. 7, show well annealed metal with equiaxed grains, a little larger than in the previous specimens.

Fig. 13 shows a typical cross section and Fig. 14 a typical longitudinal section, both magnified 250 diameters. The grains show little or no distortion; in other words, the amount of reduction after the final anneal has been very slight, if any. Season cracking would not be expected in such a structure.

CONCLUSIONS.

1. The tubes show surface flaws.

2. The metal is in an annealed condition or has had but slight final draw.

3. The grain size is good, showing that the annealing temperature was correct.

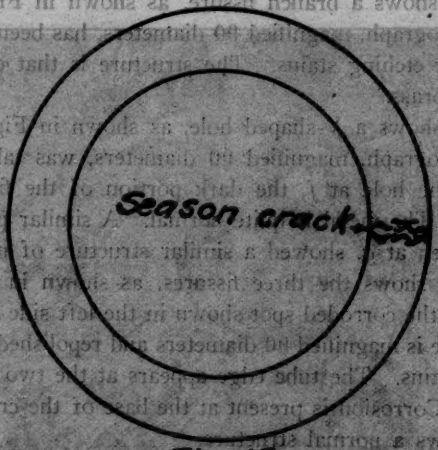


Fig. 15.

Fig. 16.

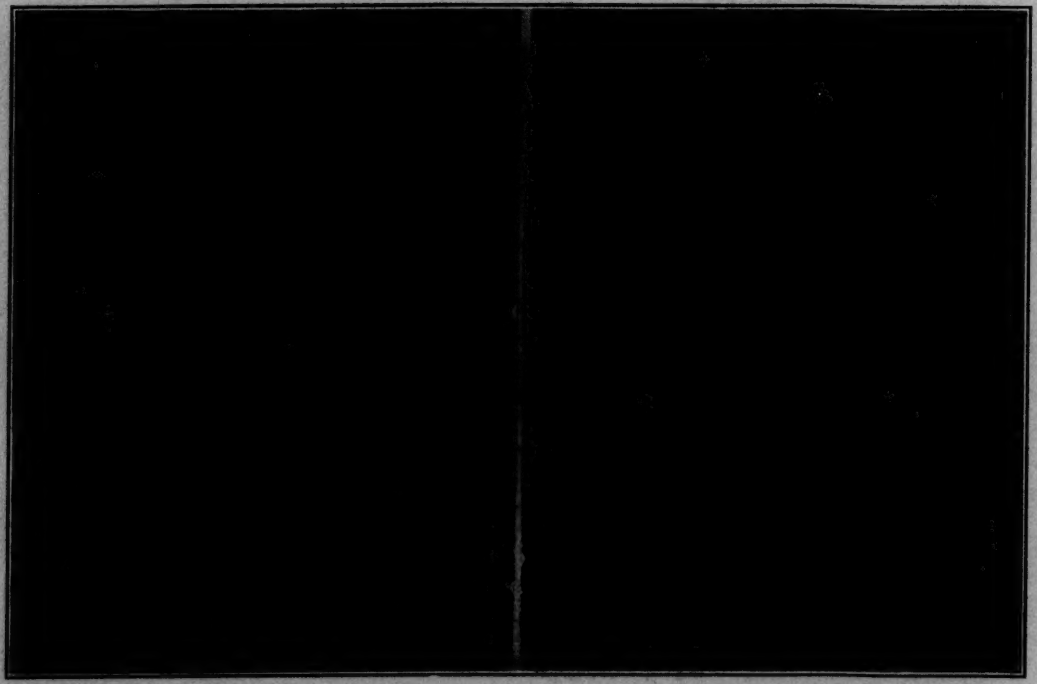


FIG. 17.

FIG. 18.

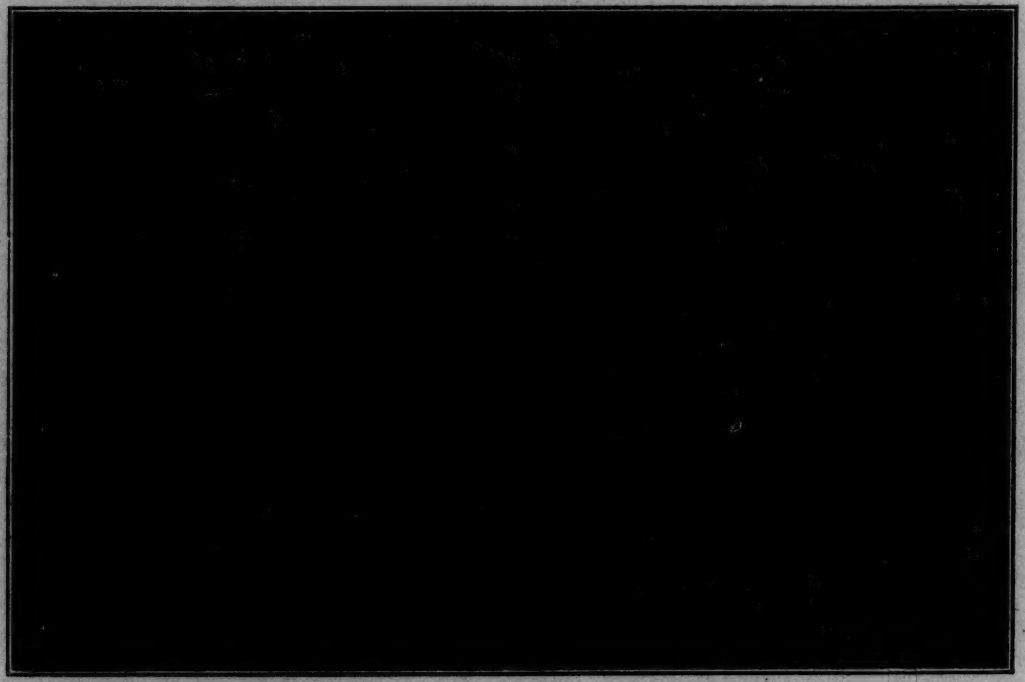


FIG. 19.

FIG. 20.

4. Corrosion started at the base of a bad seam and worked into the crack.

5. The trouble is apparently due to a tube originally faulty and not to season cracking.

CASE B.

The previous case dealt with a faulty tube, failure being due to a seam in the tube, which was composed of small, equiaxed grains. The commanding officer, in discussing previous failures of condenser tubes of similar composition, put forward the theory that they were season cracks pure and simple, and, that they occurred in cold-drawn metal.

The following case of a faulty condenser tube submitted by him for examination conclusively proves his contention with regard to this particular specimen.

Cross sections were cut through the lines of failure, and in the sound metal, Fig. 15. A longitudinal section was slightly compressed and its surface filed down and polished as shown in Fig. 16. The longitudinal section showed a number of irregular step-like cracks uniting in one main crack, in a general longitudinal direction. The cracks are intergranular.

Figs. 17, 18 and 19 are photomicrographs of the longitudinal section ($\times$90 diameters) taken at *a*, *b* and *c*, respectively, Fig. 2.

Fig. 17 shows the longitudinal section at *a*, the end of an intergranular crack. The grains are distorted by cold work.

Fig. 18 shows the section at *b*, another intergranular crack. The alpha grains show slip-lines.

Fig. 19 shows a section at *c*, with the same characteristics.

Fig. 20 shows a longtudinal section, light etched, magnified to 250 diameters. Slip lines and slip bands are very pronounced. The metal shows every evidence of cold work.

CONCLUSIONS.

1. The tube shows typical season cracking.

2. The metal was not annealed after the final draw.

3. The grains are considerably distorted and show the effects of cold working due to the final draw.

In this case the commanding officer was correct in his contentions. Thus from the same external evidence the conclusions drawn in these two cases were fifty per cent correct. The microphotograph, however, clears up the case without any question. The writer would like to see more condenser-tube failures submitted to this inspection before conclusions were drawn. It is believed that in some cases tubes of condensers are condemned as a whole as brittle or defective due to cold drawing when the only trouble is an isolated defective tube containing a seam.

OBITUARY.

GEORGE WILLIAM MAGEE.

CHIEF ENGINEER, U. S. N.

George W. Magee, Chief Engineer, U. S. N. (retired), died on the 9th of December, 1917, at East Orange, N. J. He was born in New York City on the 1st of February, 1840, and received his education in the day schools of that city.

He was apprenticed in the pattern shop of the Novelty Iron Works and became a skilful pattern maker. His marked talent for designing soon became evident, and he was retained in the draughting room.

In that day apprentices were indentured, and their employers were obligated to instruct them in each branch of the trade, including a term of at least three months in the draughting room.

George Magee was possessed in a marked degree of intuition and judgment—what the workmen used to call "gumption." To such a mind mathematics came easy, and any boy who had aptitude for drawing soon became a good designer. George Magee led in the designs of the engine for the *Constitution*, a large paddle-wheel ship for the Pacific Mail Steam Ship Company, before he was of age. His drawings were beautifully executed. There was then no school for mechanical engineers, and books on the subject had not been printed. Designers of machinery were found only among the engine builders. There was not much duplication, but engines were designed and built for special purposes, thus requiring continuous inventing and designing.

Magee designed a stationary engine having an overhung cylinder, which was such a radical departure from orthodox

practice that it was rejected with cries of "kill him," but twenty-five years later there were thousands of overhung cylinders on high-speed engines, which were making incandescent electric lighting a success.

He entered the Navy as a Third Assistant Engineer on the 4th of April, 1861, and was ordered to duty at the Washington Navy Yard, where the original machinery for the *Pensacola* was being built.

It was in the first year of the Civil War, when every available shop was being utilized for Navy work, and when the Navy Department was much occupied with inventions, designs, schemes, etc. None of them were so impossible as the engine of the *Pensacola*, which was being designed by a patent attorney and a professional inventor, jointly, and was built in opposition to the advice of the Engineer-in-Chief of the Navy.

George Magee, knowing he was to go to sea in the *Pensacola*, began at once to take notice. He promptly said it would be impossible with the hand gear to reverse that great engine in less than ten minutes, and was asked, "Well, what are you going to do about it?" His reply was to design a steam reversing engine (probably the first steam reversing engine), and, when built, the result was exactly as he had predicted.

After the *Pensacola* cruise Magee was detailed for designing work in the Bureau, where he worked over a draughting board for several years on the plans for the most successful ships of that period.

His design for the engine of the *Epervier* had the first piston valve on a Navy engine. There was an elegance and beauty in the design of that engine which has never been surpassed.

His services were considered so valuable that the Secretary of the Navy waived the required sea service in his case, and he was promoted with his class. He was commissioned a Chief Engineer in 1876.

He served on board the *Juniata*, *Roanoke*, *Minnesota*, *Tuscarora*, *Omaha*, *Ossipee* and *Miantonomoh*, a total of 14 years

and 4 months of sea service. He also served in the New York Navy Yard and at the Continental Iron Works. His last duty was on board the *Miantonomoh*, from the deck of which vessel he was washed overboard. He came near being drowned. When rescued he was so far exhausted that he afterwards said he did not know whether he was moving his hands or not. From this shock he never recovered, and was retired in June, 1893.

Personally, Chief Engineer Magee was always attractive and popular; his manners were charming. He was often able to apprehend trouble and to prevent it, and in this way he made his burdens light. He was never known to lose his temper nor to fret over misfortunes; he seemed to see most things in a rose-colored light, and was able to enjoy the fortunes which fell to him. Even in his years of sickness he took a profound interest, if not a pleasure, in meeting and in treating his maladies, and he died, as he had lived, peacefully.

He was a member of the American Society of Naval Engineers, of the Marine Engineers and Naval Architects, and of the Mechanical Engineers.—G. W. BAIRD.

NOTES.

WHAT THE COUNTRY IS FIGHTING FOR.

Extract of Address of HON. WM. H. TAFT, *Delivered at the Opening Session of the Thirty-eighth Annual Meeting of the American Society of Mechanical Engineers.*

MUST TALK ABOUT THE WAR.

I was not quite sure what I ought to talk about when I came here. What I have said up to this point, like the President's few remarks, has not been my speech. One cannot come before an intelligent audience like this and fail to talk about the war. I have talked about the war a good deal in various parts of the country, and I have been impressed with the feeling that the war could be talked about a good deal more about the country, throughout the country, to the people of the country, with very great advantage. There are so many who are doubtful—they are for the Government and they are for the war, but they are walking interrogation marks as to whether we did not make a mistake here, or did not make a mistake there, or whether we are fully and rightfully in the war, that I think an evangel ought to be preached on the subject to demonstrate that we are rightfully in the war, that we never have done anything that was not justified, and that the cause now presented to us is so righteous that if we are the people we claim to be we must win it if it costs the last man and the last dollar that we have. (Prolonged applause.)

You meet men who are now, after we have decided to go into the war, and after we are in it, who are now "judicially-minded"—that is to say, they do not say they are neutral, but they are judicially-minded. While I am in favor of being judicially-minded, I am not in favor of masquerading under a judicial mind a lack of that fine edge of loyal patriotism that we need to carry this country through the war. (Applause.) I am opposed to apathy; I am in favor of team work, and of knowing why we are in, and what we are going to do, and in favor of being determined to do it. (Applause.)

GERMANY'S BREECHES OF INTERNATIONAL LAW.

You will find this judicially-minded person suggest that we were unneutral during the three years that we were not in the war, because we furnished ammunition and other supplies to the Allies. Well, we had a right to do that, under international law. Germany herself had agreed to that rule of international law with respect to the power and duties of neutrals—not that neutral governments could furnish such supplies, but that neutral governments could permit their citizens to do so, the citizens taking the risk of confiscation of those articles as contraband if found upon the high seas. And there were those who sympathized with Germany after the German commercial marine had been driven from the seas, and they said it was unneutral for us to furnish one side.

The fortune of war was not our fault. The President was right in insisting that we should stand by the rule of international law in that regard, because if by our acquiescence the rule of international law were to be changed requiring every neutral government to suppress its citizens

from carrying on such a trade, it would only make overwhelming the advantages of a military nation that devoted itself, as Germany did for fifty years, to getting ready for this war, a nation which has piled up the ammunition and supplies needed to carry her through years of war. We, if subjected to a war of aggression, would never be ready. We would always be unprepared, and as a consequence, when forced into war, we would have to look about to prepare suddenly, and then find denied to us the right to get our material and supplies from the citizens of neutral nations, under the new rule suggested. Therefore, it would have been the wildest lunacy for us to consent to a change of international law in that regard, and the merits were wholly with the President in taking that position.

But, notwithstanding the fact that we pursued the path of neutrality as laid down by the law, Germany sank an English commercial liner having three thousand persons on board, and sent to their death one hundred and fourteen American citizens by the murderous torpedo which her submarine hurled at this vessel. Then for a year we continued a discussion, arising from Germany's unfounded claim that the vessel was armed; then she sank another vessel under similar circumstances; then we said that we would sever our relations with Germany; then she said that she would discontinue that method of warfare until further notice; then on the 31st of January last Germany notified us, as she notified the world, that she intended to resume the ruthless submarine warfare; then shortly afterwards we severed our diplomatic relations with Germany; then she sank four or five American vessels, returning to this country in ballast, and sent to their death some twenty-five or thirty American sailors; and then we declared that a state of war existed.

OUR DUTY AS A GOVERNMENT.

Now, my friends, is there anything else that we could do but that? That is where your judicially-minded person would come in. The answer to my question depends, first, on the proposition of what were our rights, and what were the rights of our citizens? and, secondly, what was our duty as a Government with respect to those rights? International law is indefinite in certain respects, but it is as definite as the law of promissory notes with reference to the rule of the capture of commercial vessels at sea. A belligerent may capture the commercial vessel of its enemy and sink that vessel. It may capture, under certain circumstances, a neutral vessel, violating a blockade, and possibly may sink it, but an incontestible rule for a hundred years has been that that right of capture and right of destruction is subject to one limitation, namely, that the ship's company of the captured vessel shall always be put in a place of safety before the vessel is sunk.

Admiral Semmes, in the Civil War, sank perhaps four hundred or five hundred vessels of the United States commercial marine, but he prided himself that in all that destruction not one single human life was lost. He was an international lawyer of repute; he was also a naval commander, and his course in that regard is the strongest evidence of what international law is on that point. Therefore Germany violated the rights of those citizens whom she exposed to death and whom she sent to death. When a man kills another deliberately, without right, it is murder, and there is no other word nor any other term in international law that can be applied to a case where a nation kills men and women and children without right. (Applause.)

GERMANY'S WARNING OF MURDER.

Ah, but it is said these people had notice. That distinguished and eminent Christian statesman, Count von Bernstorff, had whispered over the telephone and had intimated very enigmatically that any one who went

aboard the *Lusitania* would run the risk of being torpedoed, and it is stated that those who went down in ships sunk afterward knew that Germany was on the sea with these murderous instruments. Well, that is a fine plea. Suppose a man in New York should warn a neighbor that he could not go down into the street upon which his house abutted, because if he did he would kill him; and suppose this man who was warned was a courageous American citizen, who knew what his rights were and he went down into the street, and the threatener did kill him. Suppose that man was indicted and haled into court and called upon to plead, and he pleaded "Not guilty" on the ground that he had notified this man that if he would come down into the street he would kill him, and therefore he was not guilty because the man himself was guilty of contributory negligence in running into a bullet whose presence he ought to have anticipated on the street. (Applause.)

But Senator La Follette says—(hisses on the part of some persons in the audience), oh, don't hiss; it never helps to call names, no matter how poor an opinion you have of a man; it does not help the argument—but Senator La Follette says it is true they had the right to be where they were, but those were technical rights. It is too bad when a Senator in Congress, sworn to obey the Constitution, should regard the right of those poor victims on board the *Lusitania* to life and the right to protection against the invasion of a murderous nation as a technical right.

WHAT WAS OUR DUTY?

We will now assume, therefore, that this was murder of our citizens. What was our duty? The Constitution as interpreted by the Supreme Court, indeed, our general knowledge of government, would teach us that while we owe service, military and civil, to the Government, the Government owes us as a primary consideration, protection. Government is nothing but a partnership in which we are all members, and we all agree to contribute to the objects of a partnership by service and then the partnership is to help us in enabling us to enjoy our rights. Therefore, when these citizens were actually deprived of their rights, why, it is very plain that it was the business of the Government to call for reparation in respect to those whose rights had been taken away, and security and an announcement of the policy which would prevent subsequent interference with similar rights of our citizens. Otherwise, if not, then we ought to go out of the government business, because that is the object of government.

Now, Germany announced that she not only justified what she had done, but intended to continue to murder our citizens on the high seas. Our citizens are entitled to the protection of the Government at home and on the high seas and abroad. Abroad there is some qualification, because they voluntarily submit to another jurisdiction, but on the high seas, on an American vessel, and under an American flag, on that great road of the nations, they are just as much within the jurisdiction and within the protection of the Government at home as if they stood on the shores of New York, or Massachusetts, or New Jersey, and an invasion of their rights on the high seas by a foreign government is just as much an invasion as if Germany had landed a Uhlan regiment on our shores and shot into the homes of American citizens and killed them.

Therefore, if we were to continue business as a government, there was nothing else for us to do—Germany did not leave it open—except to measure swords with her in protection of those rights. If this act had been committed by Venezuela or Costa Rica, if either of those countries had sunk an American ship with a loss of one hundred lives, the President would have promptly sent a message demanding reparation and security against further invasion, and might have sent a warship down to convey

the message, just by way of suggestion, and every man, woman and child, Senator La Follette, and every pacifist, would have said, "Well done." Well, now, what is the difference between that case and the one we are considering? There is not any in principle, but there is this real difference, that Germany is the greatest military power in the world, and Venezuela is not, and therefore we are very urgently and strongly in favor of the protection of the rights of American citizens when invaded by a foreign country, provided the country is little enough, but when it is a great power—the greatest military power—then the rights are "technical." (Applause.)

Oh, my friends, there was not anything for us to do except to declare war, and a pacifist or anyone else who says otherwise or intimates otherwise, does not understand. The President has set a precedent by calling them stupid, and, after such an authority, I am willing to say I agree.

Now that brought us into the war, but when we got into the war we found what possibly we ought to have known before—some did know—that the particular cause which brought us in was only a phase of the far greater cause which the Allies were engaged in fighting. We found ourselves in the beginning ranged with democracies against autocracies. I know that our judicially-minded friend will suggest that England is a monarchy and so is Italy. Yes, that is true, but a democracy is a country in which the people rule, in which the policies of the government are determined by the popular will. The proof of the pudding is in the eating of it. Any one who knows anything about England and Italy cannot say otherwise than that the people rule in these countries, and where that is the case they are democracies. Where that is the case, the question of kings is only a question of taste. As a matter of fact, the King of Italy and the King of Great Britain have not any more to do in determining acute questions of the policies of their respective countries than an ex-President of the United States has. (Applause.)

THE MEANING OF DEMOCRACY.

Now, the President has said that we are fighting this war to make the world safe for democracy. That is a truly exact statement. But it has been misconstrued. It does not mean that we are to force democracy on other countries, that we claim to have a patent for our form of government that we are going to drive down the throats of other people. That is not what it means. It only means that the power of a people with a military and foreign policy such as that of the Imperial German Empire is dangerous to the continued and safe existence of smaller and less powerful countries that desire to have democracies and to work out the happiness of their people through that kind of government. That is what it means.

WE MUST UNDERSTAND GERMAN CHARACTER.

We cannot understand the issues at stake without understanding the character of the German people. We cannot understand their character without following their training in the last fifty or sixty years. We all have known Germans. We have liked them. When in Germany we have enjoyed seeing them. They are a kindly people; at least they were some years ago when I visited Germany. They are a kindly people, who love their homes, they love their families, they love music and they love poetry, of which they have some of the greatest exponents in the world. They conform to authority with a kind of pleasure. They are an intellectual people, they are an earnest people, a little lacking in a sense of humor, but a great people, people capable of great effort.

The truth is, while I have a profound admiration for the English people and the history of England, because having been educated as a lawyer I believe she laid the foundations of true constitutional liberty, nevertheless, I am bound to say that when I went to Europe and traveled in Europe, I would a great deal rather be closed up in a railway carriage with a German than with an Englishman; because the Englishman—I mean the regular Englishman—was constantly engaged in an affirmative effort to convince me that he did not know I was in the carriage, whereas the German was always courteous and friendly and anxious to engage in conversation.

SOME GLANCES AT GERMAN HISTORY.

The Germans for a long time were divided into twenty-eight different states, Austria the greatest of them, Prussia the next, and twenty-six others, and every one who longed for an improvement in the world, and an improvement among the Germans, wished for unity among them. There were liberty-loving Germans, and in '48 they rebelled against the Divine Right of Kings, and they had revolutions. They were not successful. They did get a constitutional monarchy for a little while in Prussia, and offered the crown to Frederick William, the great-uncle of the present Emperor, and he said he would not take it, because he got it from God, and did not purpose to take it back again out of the mud, showing that the Divine Right of Kings came honestly down that line to its present exponent.

A large number of these liberty-loving Germans were driven out of Germany and came to this country, and made one of the most valuable elements of our citizenship here (applause), and when the Civil War came on, loving liberty as they did and hating slavery, they went into the war, enlisted in great numbers, and on every battlefield in that war the blood of our German citizens was shed.

Their descendants and others who have come here since have continued to make a valuable part of our citizenship, and during these three years when we were neutral they have naturally, because of their pride in the success and prosperity of their brethren at home, had a sympathy with Germany and listened to the arguments in her behalf which have been put forth in this war. And now the war between America and Germany has come on, and their allegiance requires them to be loyal, and they are put in a sad position, and one in respect to which we should be considerate of them. But they are loyal, they have enlisted, they have gone into the draft, and contributed to the great patriotic funds and while they are not vociferous—we could hardly expect them to be so—that they are going to be loyal I have not the slightest doubt (applause), and one of the things we ought to be most thankful for is that very thing. The reason why Germany treated us as she did was because she counted on dissension among our people, growing out of the disloyality of that very element, and she has been disappointed in that regard as she has been disappointed in so many of those instances where she has attempted to read the motives of other people.

Instead of founding a constitutional monarchy, with representative institutions, and bringing about unity, as those leaders of German thought, Carl Schurz and others, hoped for, there came into the history of Germany a very different individual, Prince von Bismarck, who was the Premier of Prussia in 1862. His theory was that he would conquer and unite the German nation by blood and iron, and he developed the army, always a well-controlled body in Prussia, and he made the nation into an army, and an army into the nation, and then he planned the wars upon which he founded the unity of Germany. He became involved in a quarrel with Denmark, and induced Austria to go in with him, and took away Schles-

wig-Holstein from Denmark and then when he got it, he found it was so easy that he annexed it forcibly to Prussia and when Austria asked, in a diplomatic way, just what there had been in that war for her, he said there was not anything. And then he got into war with Austria, as he had intended, and in six week he wiped her off the map of Germany. Then in that war he annexed forcibly Hanover and Frankfort, and made an offensive and defensive alliance with Würtemberg, and Baden, and several other German countries and then he sat down to wait until that fakir, Napoleon III, in his pirouetting, would bring about an appearance of a war of aggression against Germany, which was exactly what Bismarck was waiting for, and he only had to wait four years for that and if you will read his memoirs you will see how he brought that about. You will be interested in reading, I am sure, that interview between himself and von Roon and von Moltke. They received a telegram from the Emperor outlining an interview between him and Benedetti, Napoleon's Ambassador, and they were thrown into gloom, because the interview was one which seemed so pacific to them that they thought its publication would prevent war and Bismarck sat down and, without changing the body of the message, changed a few words in it and published it; and then von Moltke said: "Now we will have war." He said: "That telegram, when it came, sounded like a parley. As you have changed it, it sounds like the rattle of a drum." This was stated by Bismarck himself. So, true to his plan, Napoleon declared war, and then, in a short time, Bismarck defeated France and took Alsace-Lorraine and an indemnity of a billion dollars, which the Germans put into the army, and Bismarck crowned a Prussian King Emperor of Germany at Versailles, and he went back to Berlin and sat down as the head of the empire to digest the pieces of territory he had bitten off in the last three wars.

BISMARCK'S SEEDS OF KULTUR.

He was not in favor of world dominion. He wanted to raise Germany to a great power in Europe, and he succeeded. He made fun of the ideal of world dominion, but there was held out to the German people the idea that all the rest of the world would try to get back from them this territory which had been taken, and therefore they must defend themselves, and so they went on and provided greater and greater armies.

They also adopted in their wonderful way, as you gentlemen of science know, the principles of science to the manufacture of everything, and to every field of industry and business. They introduced a system which they called Kultur, and which brought about a prosperity in competition with the world that attracted the admiration of the world. Their population increased and pressed upon their borders, and with their marvelous successes in the three wars, with their wonderful administration and the demonstration of their efficiency in their prosperity, and with their increase of population, they acquired megalomania, and they learned to think that they were supermen. They believed they had invented Kultur, and it was their duty to spread it over the world and enlarge their borders and conquer the world for the purposes of spreading that Kultur.

And they soon, by reason of their elevation as a people, associated themselves with God. They regarded themselves as the agents of God. They are a people of an inexorable logic. If they begin with a false premise, as they often do, their confidence in logic is such that they wipe out any fact that is inconsistent with the conclusion reached by that logic. You remember the story of the old German who was in the California gold diggings and met a man out there whom he had seen only recently in New York; and, anxious to find out how he had gotten there, he asked: "You came the plains across?" The man replied "No." "Then you have

come the Isthmus over?" "No." "Oh, then, you come the Straits of Magellan through?" "No." "Oh, I see, you were seasick coming the Horn around?" The man replied, "No," and then the German's eyes opened and he looked at the man for a minute and he said: "Well, then, you have not arrived!" (Laughter and applause.)

Having established that Kultur was necessary for the world, and that they had invented it, they believed that they were the people to spread it; and then, with that inexorable logic, all of these other conclusions followed. The state, the German state, was to spread Kultur. It was to do God's work. Therefore, every consideration must yield to the doing of that work. The state was above everything. The state, engaged in this work, could do no wrong. Therefore, these considerations of honor and decency, and the performance of obligation, could play no part. International morality was eliminated. The only sin of a state was weakness; its virtue was power. And that doctrine, or its elements, the idea that Germany was over all, was preached in the schools, in the academies, in the universities, by the great lecturers, by the military writers; and the conviction grew with the people, first, that they must protect and defend themselves and give everything to the army necessary to accomplish that, and, secondly, that they must base their state on force and maintain that force in order that they should spread Kultur to the world by domination and conquest. They eliminated, as I say, international morality.

Now, that is the nation and that is the people that we are engaged in fighting. They are obsessed as with insanity, otherwise you cannot explain what you see and read and know. "Why is it," you ask, "we did not know this before we got into the war?" Well, we read excerpts from the lectures and military writings, but we have cranks of our own—I need not mention them—but certainly we do not want to be held responsible for their writings and their statements and their actions, and we assumed that these people, thus speaking among the Germans, belonged to that necessary and conspicuous, but we hope with us unimportant, element. But it was not so in the case of Germany, and you can read now the books that have been prepared impartially showing these sermons and lectures and showing that these lectures spoke for all the people. Consider, for a moment, that there was a writer who in one of his writings incorporated a prayer like this:

"Oh, Thou, who presides over all, up above, high in the skies, up above the Cherubim and Seraphim—and the Zeppelins——"

Now, that association, if it did not shock your feelings as irreverent, would suggest a humorous view; but to the German mind, with the idea of what the Zeppelin was to do in spreaking Kultur, it was the agency of God; the association between the cherubim and seraphim, which are supposed to be God's agents, with the Zeppelins, was entirely proper. They preached sermons on the German God.

WE ARE FIGHTING THE GERMAN PEOPLE.

It is the people of Germany we are fighting, with the characteristic they have of subordination to the authority of the Prussian military regime and the Kaiser, and we must not assume they are compelled against their will to do this fighting. They have made too many heroic sacrifices in loyalty to this false idea, and in loyalty to the leadership of the Kaiser, and therefore what the President says must not be misconstrued. What we are trying to do is to separate the people of Germany from the rulers of Germany, but the only way we can separate them from their rulers is by hitting them on the head with a club so that the psychology of the situation will be brought home to them. (Great and prolonged applause.)

If you look for proof of this position of Germany with reference to the abolition of international morality you can find it in their method of war-

fare. I do not think it necessary to go into a detailed recital of the awful atrocities that have been proven before you can arrive at a general conclusion as to their violation of every rule of warfare. They bombarded unfortified towns, an act which is forbidden by international law, and the men who bombarded these unfortified towns on the east coast of England were rewarded by being decorated with the Iron Cross.

The Hague Conference provided certain rules with respect to the carrying on of war by means of aircraft, one of which was that belligerents were not to drop explosives from aircrafts on undefended towns; and the Germans promptly sent their Zeppelins, that were assembled for the purpose of carrying on war, and to which they turned for the purpose of carrying on the war, and they sent these Zeppelins to England and slaughtered innocent non-combatants. Of the thousands of victims of the Zeppelin raids, possibly not more than fifty soldiers and sailors were hit, and only one or two arsenals, but the great body of the victims is composed of women and children and old men. The men who navigated the Zeppelins in these air raids were also rewarded with Iron Crosses.

THE PATH OF KULTUR.

When the Germans entered Belgium they violated their treaties through which they had given their plighted faith for sixty years with the other nations. You would think, when they went into Belgium, under those circumstances, they would treat the people with some consideration, even in spite of their obsession. Did they? No. What they did was to take a district in Belgium and direct their soldiers to pursue the policy of Schrecklichkeit, that is, to stand up against a wall the leading citizens and shoot them, as well as the women and children. You ask for proof? Well, read the report of Lord Bryce. He is a lawyer, an able lawyer, and an historian, and he was on the committee with other lawyers and judges, and they took the evidence and dissected it and analyzed it. They rejected all the evidence as to the sporadic brutalities by soldiery which you encounter in every war, and took only the evidence of cases that could not have been committed except by the order of officers, and they showed that this was part of the military policy of Germany in terrorizing the rest of the innocent Belgians by such cruel atrocities in respect to the families of this particular little district.

But the worst thing they have done has been with respect to Armenia. When England brought over the Indian troops to help that small regular army of hers, and they came and made good soldiers, showing they had been well treated, the Germans held up their hands in holy horror and said, "They are sending Mohammedans to fight Christians," all the time having that eminently Christian monarch, the Sultan of Turkey, in alliance with them. And after the alliance was secure, then Turkey proceeded to carry out a purpose that she had partially attempted to carry out years before in ridding herself of Armenian Christians in her empire. She proceeded, with Germany looking on, and with officers of the German army at hand, through her regular soldiery and her irregular soldiery, to murder eight hundred thousand Armenians because they were Christians. Now, that is an evidence of the false philosophy, the horrid philosophy, that there is no international morality, and that nothing should stand in the way of military success and the advance of the state in the spread of kultur

That is the kind of enemy we have to fight. That is the psychological state of the German people, and the only way in which we can change it, as I say, is by defeating them. If we defeat them, then they will appreciate the falseness of a philosophy which can only be justified by victory, and then when they are defeated, as we must defeat them, then they will relegate the Kaiser—it will not need any action on our part—they will

relegate the Kaiser and the Prussian military regime to the place where they ought to go. (Applause.)

It is a very satisfactory thing to see that the sin of the Germans in this regard has found them out. When the war began, good Christians hesitated about believing in a good God, when they saw that so many innocent men could be hurled into a vortex of destruction, agony, suffering and death like this. Now the thing is cleared away, and what we see is that the world has been suffering from a cancer of militarism, and Germany has been responsible for it, and she has led the world on to these great armies, and on her hands is the blood of this awful war—this war with fifteen or twenty times the number of men engaged in it, and with an equally increased amount of suffering and agony, compared with any other war—with 40,000,000 men engaged, 7,000,000 men dead, 6,000,000 men in the hospitals, and 6,000,000 men in prison camps. That is due to Germany. The causes cannot be cut out but by suffering. God works by inexorable laws, and the penalty of sin must be paid.

This is a German war of aggression, as any schoolboy can now see. The White Paper did not show any communications between Germany and Austria during that anxious time, and they have never been disclosed, but we know now that Russia was not prepared, and England not any more prepared than we are today, and France was very lacking in her preparation, and yet we are to believe that these three countries conspired to attack Germany who was ready to the last cannon and the last reservist. Why, that is enough to make a horse laugh. It is true that Germany did not advise the killing of the Crown Prince and his consort. That is not the way Germany has begun her wars. She gets ready. She plans a war. She gets ready, and then she waits for the opportunity so that it shall seem to be a war of aggression by other powers. That is true in every war she has waged since Prussia has been in power.

GERMANY'S WOEFUL BLUNDERS.

So to go back to this sin of Germany finding her out. She has been perfect in military preparation; she has been perfect in military strategy, but where has she made her blunders? She has made her blunders, and her great blunders, in misreading other peoples, in her diplomacy, and she has made these blunders because she has eliminated from her own soul considerations of morality and motives of good, motives of service and allegiance and unselfishness, and therefore she has eliminated those from her judgment when she goes to judge of what other people will do. So she made a mistake about Great Britain, and her conscience in respect to Belgium. She made a mistake as to the British possessions—I mean those independent dominions—she said, "The tie which binds the dominions to the mother country is very light. There is no reason why they should go in, there is nothing in it for them," and she was indignant and exasperated when she found that her judgment in that regard was wrong. That is because she could not appreciate the filial relation between those countries and Great Britain. She could not appreciate the daughter's loyalty to her mother that had protected her. Is there anything more noble in this world war than the way in which Canada and Australia have responded to the call of the mother country? Canada has sent 420,000 men, and Australia 400,000 men, Australia having a population of five millions and Canada six or seven millions. In proportion, we would have to send an army of seven millions. And then France. Germany said France was decadent, permeated with socialism, no patriotism, and deeply affected with frivolity. France was not prepared, but she rallied her legions, and is it not inspiring to think of the fight that she made, knowing that the German military staff was attempting to crush France first, and she stood up, and with that thin line of the British regular army, she hurled back the German hordes at the Marne and saved the world. (Great applause.)

The biggest mistake she has made has been with respect to this country. I remember some of the things the papers said—they said we were a tangoing nation. They said we were too fat to go into the trenches. They had a contempt for us because we had not prepared for war. They assumed our citizenship of German origin would prevent the war, and political considerations would divide the people in that regard. They were also obsessed with the idea that they could end the war with this murderous weapon, this weapon they could not use except by accompanying its use with the murder of neutral people. So they went in.

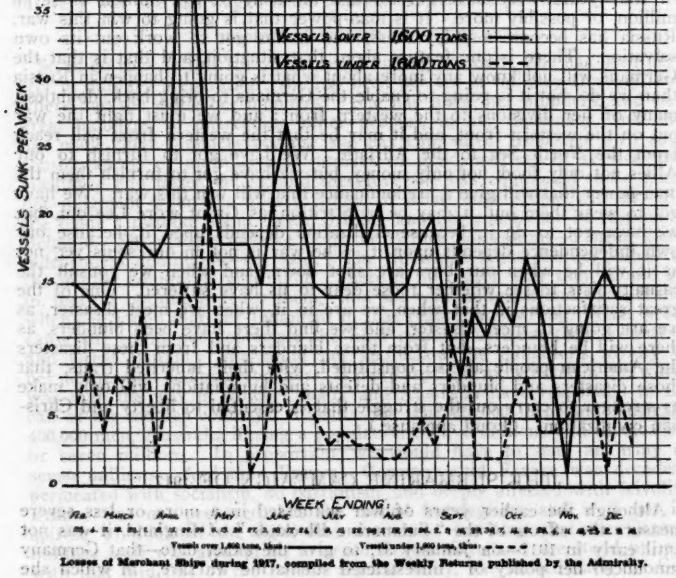
Now, ten, fifteen or twenty years hence, when our grandchildren go to their fathers, after having read a history of this war, they will say: "Papa, why in the name of common sense did Germany force the United States into this war?" And papa will have a hard time to tell, unless he goes into all of the circumstances and treats the subject from a psychological standpoint, because the boy will say, any child would say: "Why they had been fighting this war for three years, exhausting as no other war has been before, so that they were all not exhausted, but nearly so, and at that time they deliberately forced into the war against them that gigantic young nation that could furnish what was absolutely necessary, and what must determine victory in the war—more food, more money, more fighting men, than any nation in the world."

Now, that is what they have done, and nothing can explain it except the obsession that I have referred to—their failure to see things in other people, because they have eliminated from their own consideration those moral motives. Now what are we going to do about it? I have said, potentially, we are the greatest power in the world. We are a potential military power, and we have got to make that thing which is potential actual, and that is no mean job. We have before us a war of two, three or four years. We have got to raise an army of five million or seven million, or possibly more. It is man-power that is going to win this war. Russia has become a pulpy mass, and it has got to work out its own salvation. There is one feature about that situation, and that is that the Germans will not know any more about what is going to happen in Russia than we do, but it is going to enable the Germans to bring back, doubtless, many of her divisions to the western front; and we must fight the war out on the western front, and it may be that the western front will reach from the North Sea to the Adriatic. We have got to furnish to our Allies not only food, not only money, but we have got to furnish them the man-power that will give a predominance that will win this war. We have got to wear them out, it may be by attrition, as Grant wore Lee out, but we have got to do it, because civilization depends upon it, because our own independence depends upon it. The war is not in our souls yet, not as it will be when our boys are shot down, and when we consult the casualty lists to see whether those dear to us have suffered. One of the great satisfactions is that when we are in it, when we meet disaster, as we are going to meet disaster, and we find there have been blunders, as there will be blunders—but from these blunders and from these disasters the American people are so constituted, with their inherited traits, that those disasters and blunders and defeats and humiliations will only make us stronger to carry out the struggle that is essential to liberty and Christian civiilization. (Loud applause.)

THE SUBMARINE CAMPAIGN OF 1917.

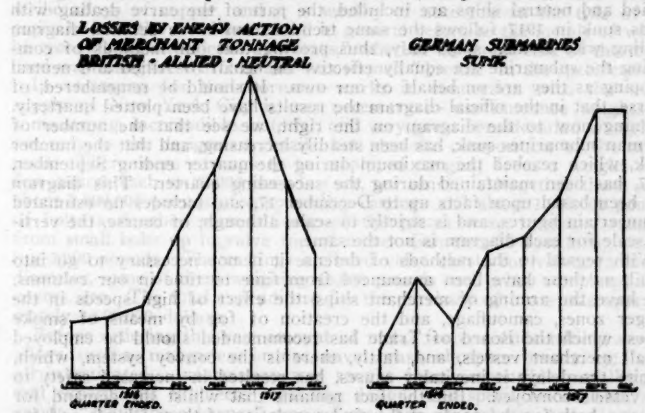
Although the earlier years of war witnessed in a more or less severe measure the effects of the "Submarine Blockade" of England, it was not until early in 1917—on January 31, to give the exact date—that Germany announced her policy of "unrestricted submarine warfare," in which she

expressed the intention of sinking all ships, whether British, Allied or neutral, which entered into the so-called "danger zones." These danger zones were explicitly defined and have latterly been extended, although their existence did not make much difference to the safety of a vessel which expected immunity from attack by sailing without the danger zone, and it must be admitted that the measure of success which attended the efforts of the pirates was such as to cause considerable apprehension as to our ability to withstand the grievous losses of merchant ships which resulted from the intensified action of the enemy submarines. The Admiralty, in answer to the popular demand, began to publish at the end of February, a weekly summary of our losses, which, whilst they did not give the actual tonnage of ships sunk by the enemy, they gave the actual number of British ships sunk per week expressed in two categories, viz: those *over* 1,600 tons and those *under* 1,600 tons, respectively, and as these results appeared from week to week the public were able to form some idea of the extent of our losses. But in order to completely realize the success or failure of the campaign and *per contra* the failure or success of our measures to combat it, it is necessary to be able to see the whole of the year's figures at a glance, and we have therefore plotted the figures as they were published week by week, thus obtaining two graphs, the first of which, represented by the full line, shows the number of vessels over 1,600 tons sunk each week throughout the past year, and the second, represented by the broken line, which shows the number of vessels under 1,600 tons sunk each week during the same period.



Losses of Merchant Ships during 1917, compiled from the Weekly Returns published by the Admiralty.

From a study of the graphs it will be seen that, with the inevitable fluctuations, from the outset a considerable measure of success attended the efforts of the enemy, and that these successes reached the zenith towards the end of April. Thus for the week ending April 22 there were 40 British vessels of over 1,600 tons sunk, and 15 vessels under 1,600 tons sunk; a number which, had it been possible for the enemy to continue to repeat would have quickly spelt national disaster. That, however, was the blackest period of the whole campaign, and subsequently matters brightened, so that, as shown by the general trend of the diagrams, the number of vessels sunk gradually decreased, until, for the week ending November 11, there was only one vessel of over 1,600 tons sunk and five under 1,600 tons. Since then the results have not been quite so gratifying, the latter weeks of November yielding an ominous "peak" to the curves; but in December the lines are again on the downward direction and the results for the week ending at midnight, December 22—not shown on the diagram —show a welcome improvement on those for the previous week.



LOSSES OF MERCHANT SHIPS AND ENEMY SUBMARINES SUNK DURING 1916 AND 1917.

A closer study of the diagram reveals the interesting fact that with regard to vessels over 1,600 tons there is a regular periodic recurrence of peaks, *i. e.*, weeks of high losses. These do not—as many writers have stated—coincide with periods of full moon, when it might be expected that, being able to operate at night as well as at day, the enemy's efforts would meet with more success; but it appears as though at intervals of six or seven weeks the number of submarines operating is a maximum, thus suggesting that this is the approximate period of each cruise, and that at the end of that time the U-boats are forced to return for a similar period to their bases for a renewal of supplies, so that with two distinct fleets operating the time of greatest activity is when one fleet is just commencing its cruise and the other has just completed it. A study of the lower curve shows that whilst the fluctuations are more erratic, suggesting that the enemy concentrates his attention more upon the larger vessels, the same periodic activity is noticeable. Taken both together, it can, we think, be safely stated that whilst the menace is serious enough to demand the continuation of every effort calculated to defeat it, the diagram shows that

our counter-measures are meeting with an ever-increasing measure of success, and that the enemy's boasted "submarine campaign," which was publicly announced as calculated to bring us to our knees in August, 1917, is not only held, but gradually being mastered.

A word as to the methods adopted to deal with the menace will not, we think, be out of place here. The methods adopted may be classified under two headings: (1) Offensive; (2) Defensive. The first consists of seeking out and destroying the submarines. Of the details of this method we can, of course, say nothing, that is the work of the "Silent Navy," but when the time comes for the veil to be lifted, the public will realize that the navy has shown a fertile activity which is in complete harmony with its high traditions. The results of its efforts are, however, graphically shown on a diagram to which Mr. Lloyd George drew attention in his speech in the House of Commons on December 20, and reproduced above. On the left-hand side of the diagram we have the losses of merchant ships—British, Allied and neutral—and it will be noticed that although the Allied and neutral ships are included, the part of the curve dealing with ships sunk in 1917 follows the same trend as our more detailed diagram dealing with British ships only, thus proving that our methods of combating the submarine are equally effective on behalf of Allied and neutral shipping as they are on behalf of our own. It should be remembered, of course, that in the official diagram the results have been plotted quarterly. Turning now to the diagram on the right, we see that the number of German submarines sunk, has been steadily increasing, and that the number sunk, which reached the maximum during the quarter ending September, 1917, has been maintained during the succeeding quarter. This diagram has been based upon facts up to December 17, and includes no estimated or uncertain figures, and is strictly to scale, although, of course, the vertical scale for each diagram is not the same.

With regard to the methods of defense it is not necessary to go into detail, as these have been announced from time to time in our columns. We have the arming of merchant ships, the effect of high speeds in the danger zones, camouflage, and the creation of fog by means of smoke boxes, which the Board of Trade has recommended should be employed by all merchant vessels, and, lastly, there is the convoy system, which, despite the delays it inevitably causes, has resulted in increased safety to the vessels convoyed. But the fact remains that whilst the demand for tonnage, both for the needs of the civil population of these islands and for the needs of our fleet and our armies abroad, continues to grow, ships are still being sunk, and so the need for new ships, far from becoming less, tends to become more and more urgent. If 1917 was a year of strenuous activity in our shipbuilding yards, it is certain that 1918 must be even more strenuous, so that we may satisfy the demands for "ships, more ships, and still more ships."—"Shipbuilding and Shipping Record."

BRASS AND OTHER COPPER ALLOYS USED IN MARINE ENGINEERING.

By Mr. J. T. Milton (Vice President).
Institute of Marine Engineers.

Introductory Contribution by the Author for the Adjourned Discussion on Tuesday, 9th October, at 7 P. M.

Since reading the paper on the above subject my attention has been drawn by one of its authors to a paper upon the inspection of brass and bronze, presented in June last to a meeting in America of the American

Society for Testing Materials by Messrs. Alfred D. Flinn and Ernst Jonson. Also to two papers by Mr. Flinn upon experiences with brass in civil engineering works. One of these latter was read at a meeting of the Municipal Engineers of the City of New York in November, 1914, the other at a meeting of the American Institute of Metals in October, 1915. Mr. Flinn is deputy chief engineer of the Board of Water Supply of the City of New York, and Mr. Jonson is an engineer inspector of the same board. It is considered that the papers are the most remarkable and most valuable of any which have been published on the subject of "Brass," and well worthy of being brought before this Institute.

Both of these latter papers contain descriptions of the brasswork used in the construction of the new water works of New York City, commonly referred to as the Catskill Aqueduct, and they record some of the experiences with the brass from 1908 onwards.

Mr. Flinn states that in America, as is the case here also, considerable confusion exists as to the use of the terms "brass" and "bronze" to describe alloys of copper, and it is worthy of note that in the title of his paper he uses the word *brass*, and in general refers to the metals of which he recounts the experience as "brass," whereas the great bulk of the metal used was manganese bronze and was termed "bronze" in the specifications.

In arranging for the enormous works in connection with the water supply of New York City it was necessary to use immense weights of some strong, incorrodible material, and very great consideration was given not only to the designs of details, but also the composition of the metal used. There were castings to be employed ranging from a weight of a few ounces each to 22,000 pounds (say, ten tons). Altogether there were about 2,000,000 pounds of copper alloy castings and about 1,000,000 pounds of bolts and other parts. Some of the pieces were forgings, ranging from small bolts up to valve rods 6 inches diameter, 31 feet long, weighing 3,200 pounds. The castings included valve chests for valves 72 inches diameter, some of them weighing 44,000 pounds each when completed with covers, etc. The total expenditure for the purchase of brass was approximately 1,000,000 dollars.

Before ordering this vast amount of metal for one undertaking, several alloys, such as manganese bronze, naval brass, phosphor bronze, etc., were all investigated. Information was sought from responsible metallurgists and experienced engineers regarding designs and specifications, and, as a result, "manganese bronze" was selected for general use, but not to the exclusion of all other alloys. All the articles were obtained under specification and were carefully inspected. Most of the work was carried out by very experienced manufacturers by their own methods, some of the firms having had experience for a very large number of years. Whenever possible hydrostatic and other tests were made, in addition to the customary tensile and bend tests. The specifications stipulated the desired physical qualities, but not methods of production nor metallurgical composition.

The following extracts from the specifications are taken from Mr. Flinn's papers:

Bronze Castings.—All bronze castings shall be made of new metal, shall be free from objectionable imperfections, and shall conform accurately to patterns. When the castings are being machined, if the metal shows signs of imperfect mixing they shall be rejected. Unless otherwise called for in the specification, or on the drawings, bronze where indicated upon the drawings shall mean manganese bronze.

Manganese Bronze.—All manganese bronze shall be equal to Spare's Parson's or Hyde's manganese bronze, and shall have a tensile strength of not less than 65,000 pounds per square inch, an elastic limit of not less than 45 per cent of the ultimate tensile strength and an elongation of not less than 25 per cent in two inches.

Brass Rivet Rod.—Tensile strength of brass rivet rods shall be not less than 55,000 pounds per square inch. The elastic limit shall be not less than 30,000 pounds per square inch and the elongation not less than 20 per cent.

Stems.—The main gate stem shall be of manganese bronze or other bronze of approved composition, of such dimensions that when placed in tension under a load in pounds, equal to the area of the valve opening in square inches, multiplied by 125, the minimum cross section of the stem shall have a resultant unit stress not exceeding two-thirds the elastic limit of the material used. These stems shall be turned straight and true, and shall have all threads lathe-cut.

Rolled Bronze.—Whenever the term bronze is used in these specifications in a general way or on the drawings without qualification it shall mean manganese or vanadium bronze or monel metal. Whenever the characteristics of any material are not particularly specified such material shall be used as is customary in first-class work of the nature for which the material is employed.

TYPICAL RESULTS OF PHYSICAL TESTS OF BRASSES USED ON THE CATSKILL AQUEDUCT.

Yield Point lbs. sq. in.	Ultimate Strength lbs. sq. in.	Ratio of Yield to Net.	Elongation per cent.	Reduction of area per cent.	Fracture.
Forgings.					
36,500	73,150	50	41.5	46.8	Irregular.
37,500	75,750	50	35.5	43.9	Irregular.
38,250	76,900	50	35.5	46.8	Irregular.
52,500	77,100	68	31.0	...	Irregular Silky.
49,300	76,150	54	33.5	...	Silky Cup.
50,000	75,350	67	31.0	...	Irregular Silky.
43,500	70,000	62	34.0	47.0	Irregular.
36,000	67,500	53	40.5	43.5	Irregular.
Castings.					
39,650	66,000	60	40.0	...	Irregular Grain.
43,100	68,850	53	25.0	...	Irregular Grain.
39,950	68,050	59	29.0	...	Irregular Grain.
33,000	66,250	50	25.0	25.4	Irregular.
32,750	67,500	48	30.0	27.7	Irregular.
32,250	67,200	48	36.0	34.1	Irregular.
39,500	69,100	57	39.0	...	Irregular Silky.
47,500	71,000	67	40.5	...	Irregular Gran.
50,000	70,000	71	34.0	...	Ang. Silky.
33,250	73,000	45	30.0	28.5	Irregular.
33,000	73,500	45	35.0	34.0	Irregular.
34,500	73,250	47	40.0	36.5	Irregular.
43,250	69,950	62	40.5	...	Ang. Silky.
44,000	70,050	63	34.0	...	Irregular Cup.
42,500	69,500	61	37.5	...	Ragged Cup.
46,000	69,850	66	33.0	...	Ang. Cup.
44,000	69,600	63	40.5	...	Ang. Fine Gran.
41,500	72,350	57	36.5	...	Irred. ½-Cup.
40,000	68,000	59	38.0	...	Forked Silky.
42,000	67,000	63	37.0	...	Ang. Silky.
43,750	69,050	63	37.0	...	Forked Silky.

The minimum physical properties of bronze shall, except as otherwise specified, be as follows:

Castings—
Ultimate tensile strength, 65,000 pounds square inch.
Yield point, 32,000 pounds square inch.
Elongation, 25 per cent.

Rolled material, thickness one inch and below—
Ultimate strength, 72,000 pounds square inch.
Yield point, 36,000 pounds per square inch.
Elongation, 28 per cent.

Rolled material, thickness above one inch—
Ultimate strength, 70,000 pounds square inch.
Yield point, 35,000 pounds square inch.
Elongation, 28 per cent.

After being forged into a bar, rolled or forged bronze shall stand: First, hammering hot to a fine point. Second, bending cold through an angle of 120 degrees to a radius equal to the thickness of the bar.

Bronze Pipes.—The pipe shall be seamless-drawn, semi-annealed, iron-pipe-size tubing of the sizes shown on the contract drawings or ordered. It shall be made of the bronze specified in the general sections or of an approved commercial bronze. All pipe furnished under this item shall be free from season cracks, surface cracks, or other defects.

When the pipe is finished, ready for shipment, the engineer will subject about 1 per cent of the lot, taken at random, to the following tests:

1st.—Each test pipe shall stand threading perfectly, with a die with the usual thread for the size of pipe.

2d.—After annealing, the end of each test pipe shall stand being flattened by hammering until the sides are brought parallel with a curve on the inside of the ends not greater than twice the thickness of the metal, without showing cracks or flaws.

3d.—After annealing, each test piece shall have a piece 3 inches long cut from it, which, when split, shall stand opening out flat without showing cracks or flaws.

Mr. Flinn states that, in spite of the care exercised, the brass furnished proved to be distinctly unsatisfactory in most of the uses to which it was put. No suspicions of definite troubles developed until late in 1913, when numerous bolts and rods were found to be cracked, and since early in 1915 some of the castings from at least three different foundries have also been found to be cracked. All these castings had, before acceptance, been subjected to a hydrostatic pressure of 200 and 300 pounds per square inch for a half hour or more, yet some months later, after having been erected in position, they leaked under pressure of only a few pounds. In some castings the cracks have extended since they were first observed, and in some additional cracks have been discovered as time has elapsed. In nearly all of the cases of the cracked castings the defects appeared to be close to or in a repair made by "burning."

In the case of defects in plates, bolts, rods, sides, and rungs of ladders, etc., some of the articles had not been taken out of the packing cases. In some cases the cracks were fine and superficial, but in others they were deep. In some cases in bolts from ½ inch to 2¼ inches in diameter the whole cross section was affected, some of the bolts being found to be severed. Two or three years have passed in some cases before the defects developed sufficiently to be detected.

Mr. Flinn states that but for the seriousness of the matter "bronze" or "brass" would be but a laughing-stock among the engineers of the aqueduct.

In the 1914 paper Mr. Flinn mentions that only a few small brass pipes had failed. He says, indeed, that there is but small excuse for supplying other than dependable brass pipe nowadays, as correct methods of manu-

facture are well known in the trade; but in the 1915 paper he states that serious failures had been experienced with pipes also.

He states that the defects were found to be so general and so distributed through the output of various manufacturers, and were so common in the different kinds of brass that all wrought brass fell under suspicion. After the first discovery of the crackings, replacements were made with brass produced by methods avoiding cold working in the hope that by this further trouble would be avoided, but unfortunately this did not prove to be the case.

It was well known that cold working produced initial strains in the material, and after the discovery of the defects it was decided to use plain extended or hot-rolled rods wherever practicable and to anneal all material which had to be drawn or rolled cold, in the hope that further trouble would not occur, but, as stated, this hope was not realized, as plain extended, hot-forged and annealed brass rods, supposedly free from initial strains, have also failed in disturbingly large quantities.

It is noteworthy that at the time of reading the paper in 1915 Mr. Flinn was able to state that large forgings, which include chiefly the items for large sluice gates and valves, have shown no sign whatever of failure. He states: No brand or make of brass or bronze has wholly escaped. Manganese bronze, naval brass (including a well-known bronze and its imitation), and Muntz metal from all the manufacturers who have furnished any considerable quantity, all have failed.

Mr. Flinn makes some very drastic remarks as to the want of such knowledge of the physical characters and capacities of this group of alloys as is required to be a safe and dependable guide to their manufacture, inspection and use. As his investigations and experience progressed he found not only unfortunate foundry and shop ignorance of important details of the manufacture of copper alloys, but also seemingly a lack of definite and complete knowledge of some of the fundamental characteristics of these alloys. He says: Before civil and mechanical engineers can venture to use brass extensively for important works where it would be subjected to other than very low stress, some questions will have to be conclusively answered by manufacturers and experts. The following are the questions he submits:

Can brass or bronze of high tensile strength be reliably produced which can be used safely for important permanent structures in such parts as bolts and other rolled, drawn, extruded or forged shapes?

What should be the specifications for such brasses or bronzes?

What inspection methods and tests should be used?

By what tests can the tendency to subsequent failure be detected at any time after manufacture?

What working stresses can be safely used for these various alloys?

Will these brasses or bronzes deteriorate by reason of constantly applied or frequently repeated stress—*i. e.*, will they fail from fatigue?

Can large hollow manganese-bronze castings be made of such forms as water and steam valves?

If manganese bronze is not suitable for such castings, what composition can be used?

Can any repairs of brass castings to be subjected to hydrostatic pressure be safely made by any process of "burning in" or welding now commonly used? If so, what relation has the shape and size of the castings to the methods and means to be employed?

What tests can be applied to prove that such a repair has been successfully made, and that later a crack will not develop because of it?

Mr. Jonson made some experiments to show that cold-worked brass was always in a state of initial stress. He subsequently found that very similar methods of test had been adopted by Professors Heyn and Martens in

Germany. An account of these latter was given by Professor Heyn in his May lecture to the Institute of Metals in 1914. Mr. Jonson in one set of experiments took 6-inch lengths of round rods, faced their ends into parallelism and accurately measured their length. He then reduced the diameter of 4 inches of the length by successive steps of ⅛ inch, measuring the length of the turned specimen. This became longer at each turning, until a small central core only was left. The experiments showed that initially the outer parts of the piece were in tension and the center in compression. As the portions in tension were successively turned off, the compression became somewhat relieved, and the piece consequently elongated.

Messrs. Martens and Heyn developed a method for determining the tendency of brass to crack by immersing specimens in a solution of a mercury salt. In much-strained samples cracks would then occur in a short time, but in other samples, presumably not so much stressed, the cracks would occasionally not so much stressed, the cracks would occasionally not become apparent for a number of days. This method was found to be useful, but not infallible, as some rods which within a reasonable period of observation developed no cracks by the application of the mercury solution subsequently spontaneously cracked.

This method of test was thought to depend upon the corrosion of the surface, and many experiences indicated that corrosion had much to do with the cracking under some circumstances. This was amply proved by some experiments upon samples, which were subjected to considerable strain and then submitted to corrosive influences. Some pieces of old corroded brass rods were also tested. They would stand bending only 45 degrees to 80 degrees before fracture, but other pieces of the same rod, when cleaned with emery cloth could be bent through 160 degrees, or even flat on themselves, without breaking. On the other hand, to show that corrosion is not a necessary factor in the cracking of brass, it is only necessary to recall that many specimens were found to be cracked when they were removed from the packing cases before being placed in use.

Mr. Flinn, in both papers, states that the trouble experienced may be classified as follows:

1. Breaks from stress, (*a*) initial stress due to methods of manufacture and (*b*) applied stress due to use.
2. Damage by wrong heat treatment, as in forging, bending, flanging, upsetting and annealing.

Damage of Class 2 results entirely from lack of skill, knowledge or care on the part of the manufacturer.

Damage of Class 1 (*b*) is dealt with in the joint paper of Messrs. Flinn and Jonson. They point out that manufacturers claimed for some of their special bronzes greater strength and toughness than steel possesses, besides being so much less liable to deterioration from corrosion. Ultimate strength and ultimate extension, however, are not the qualities which should determine the greatest stress which might be used in structures; this should be determined by the yield point of the material, the ultimate strength having little to do with the safe working stress when comparing different sorts of material.

The authors, however, are careful to refer to the "true" yield point of the metal, and state that a high yield point signifies nothing unless it is known what portion of this yield point is cancelled by initial stress, and, therefore, not available for working stress. They point out that if a test piece is continuously stretched by a gradually increasing load, when the true yield point is reached the metal flows until it is sufficiently hardened to carry the load. This, of course, is well known to be so in the case of all ductile materials. If a piece of ductile metal, whether it is a test piece or a piece of structural material, is subjected to a stress which deforms it,

its yield point as regards further deformation due to the same kind of stress is raised, and it will not further deform until it is subjected to a greater stress than that initially put upon it. When it is seen that in the tests of the material used in the Aqueduct work quoted in Mr. Flinn's papers that the ratio of yield point to ultimate strength ranges from 50 to 68 per cent in the case of forgings and from 45 to 71 per cent in castings, one is forced to conclude that some of the results were obtained from specimens which were either subjected to high initial stresses or which had been submitted to high stress before test, and that, therefore, the recorded yield points were not "true." It is precisely this difficulty which has led our engineers to omit conditions as to yield point from acceptance specifications for ordinary steel for constructive purposes.

The authors of the paper upon "The Inspection of Brass and Bronze" consider that acceptance tests should include measurements of initial stress, and they suggest methods for determining this. They state that initial stress may be relieved in two ways: either by stretching the metal near the surface by mechanical work or by temporarily reducing its elastic limit by heating, and thus allowing it to be stretched by the compressive stress of the interior of the metal. The mechanical working used for the purpose is either squeezing between the rolls of a straightening machine or bending the bar successively in four directions at right angles to one another. This operation in America is given the name of "Springing." Annealing, however, appears to be the more reliable method. They state that it has been proved that if sufficient time is allowed for annealing the temperature need not be so high as is required where initial stress has to be eliminated by quick annealing so that it is possible to eliminate initial stress by slow annealing without appreciably lowering the yield point given by cold drawing.

The question of "burning on" tips of propeller blades was referred to when the previous paper on brass was read at this Institute. In this connection the remarks of Messrs. Flinn and Jonson upon foundry repairs will be of considerable interest.

They say that as brass and bronze castings are expensive to make, it is not desirable to reject such castings on account of defects which can be remedied. Minor leaks in a casting may be stopped by peening, which, however, they consider to be a questionable method of treatment. For very small defects they prefer to drill, and then to screw in a plug of the same metal as the casting. If this is not practicable, the defective portions should be cut out by chisel or drill till all the defective metal is removed and the cavity then filled by pouring in metal from a crucible in such a way that the entire surface of the cavity is melted, and then consolidated by the added metal. If a gas flame is used for this there is a risk of the filling not being solid, besides the possibility of oxidation. When this method is adopted the cooling of the metal will be accompanied by contraction, which sets up severe tensile stresses in the added metal and its surroundings. Further, the rapid cooling of the added metal makes its elastic limit higher than that of the neighboring portions. The state of tension so set up may be so severe that after a time cracks will appear.

The stresses may be lessened by keeping the casting heated to a high temperature while the repair is being made, but the best way to prevent cracking is to anneal the casting immediately the repair has been made. The annealing temperature should be maintained for several hours, so as to give the metal time to flow. This, they say, is the only way to insure safety in castings in which "burning in" has been adopted.

In the "Engineering Record" of 12th August, 1914, there appeared a communication from Mr. L. D. Van Aken, the Superintendent of the National Brass and Copper Tube Company, of Hastings-on-Hudson, New York, on the subject of the cracking of brass and bronze. He states

that the frequency of these cracks has alarmed metallurgists, engineers, etc., and he attributes the failures mainly to what he terms the greed for higher tensile strength, which has led to a disregard for safety. He mentions that the U. S. Navy has adopted reasonable specifications for the elimination of the trouble. He records some experiments which he had made upon a brass having a composition of copper 61.33, zinc 37.66, lead 0.16, tin 0.83, iron 0.02. (This is very nearly the composition of our well-known naval brass.) This was first made into an extruded rod, 1⅛-inch diameter, allowed to cool slowly in the air and then tested. The result is given as No. 1. The rod was then cold drawn to 1 inch diameter, with the result given as No. 2. After this the cold-drawn rod was annealed at 1,300 degrees F. for 30 minutes, and then allowed to air cool; the results of tests in this condition are given as No. 3. A portion of the rod after the same annealing was quenched in cold water instead of being allowed to cool in the air; the results are given in No. 4. Mr. Van Aken states that the annealing as in No. 3 is sufficient to prevent crackage of any description.

No.	Ultimate strength lbs. per sq. in.	Elastic limit lbs. per. sq. in.	Ratio of Elastic to Ultimate per cent.	Elongation in 2 ins. per cent.	Reduction of Area per cent.	Hardness Scleroscope.
1	58,000	26,550	46	57.5	59.5	23
2	73,100	70,100	96	27	45.2	29½, Surface 53
3	58,000	27,200	47	53	81.5	20
4	63,800	29,900	47	38	38.5	20

It will be noticed how enormously the cold drawing has affected the tensile strength, and more especially the elastic limit, and that the cold-worked material still possesses as much ductility as measured by the extension and by the reduction area as would be shown by a sample of a mild-steel forging. Judged from the tensile test, it might be thought to be an excellent constructive material. Its high tensile strength, however, is dearly bought if it is accompanied by the tendency to spontaneously crack. The tests show also that annealing has restored the qualities of the metal to their original amounts.

It is to be noted that Mr. Van Aken only tested extended rods. In Mr. Flinn's papers extended rods are mentioned in such a way as to lead to the inference that extruded metal may me received with the same confidence as rolled or forged material. However, no comparative tests are given of the results of the two methods of manufacture. It is extremely desirable that some exhaustive experiments should be undertaken with different compositions of metals to show the effect upon the metal of the different processes, viz: casting pure and simple, forging, hot rolling, extrusion and also cold work, say, by drawing. These tests should include the effects of annealing. They should also include the resistance to repeated applications of stress, commonly called fatigue tests. In fact, this kind of test is of greater importance than any other. It is evident that at present there is a lack of knowledge of the physical properties of many of the much-used special alloys, although we have been using them for many years for very important structural purposes.

In the same volume of Transactions of the American Institute of Metals, which contains Mr. Flinn's paper, there is another communication by

Dr. P. D. Merica and Mr. R. W. Woodward upon "The Failure of Structural Brasses." Dr. Merica is Physicist of the Bureau of Standards, Washington.

In this paper there is a description of some work done in the Bureau of Standards in investigating the causes of spontaneous cracking in brass. The authors consider that failures are due primarily to the presence of internal stresses, and state that manufacturers should endeavor to produce material free from these stresses, and yet possessing the physical properties produced by the mechanical treatment which is responsible for these stresses. They state that it is possible, for instance, to make naval brass practically free from internal stress and yet having an ultimate strength of 60,000 pounds and an elastic limit of from 30,000 to 35,000 pounds per square inch.

The paper gives the results of numerous tests made with samples, some of which were new material, and others of brass which had been put into service. The authors tested these samples for internal stresses by Heyn's method. They state that their results may be relied upon to have a maximum error of not more than 1,000 to 2,000 pounds per square inch, or 5 per cent.

The figures they give show that all the samples they tested were more or less initially stressed, some of them to an enormous extent. Some of the samples have tension stresses on the surface, and others compression stresses.

They investigated the method of testing for initial stress by immersing the samples in solutions such as mercuric chloride, mercurous nitrate, ammonium hydroxide, etc. Cracks were often produced, sometimes in two minutes, sometimes in 24 hours. They found, however, that while those samples which had a high tension on the surface cracked, those which had large compressive stresses did not. This test, therefore, is not a sufficient one to enable defective material to be detected.

The following are the main conclusions arrived at by the authors:

1. Failures occur, so far as is known, only in brass which has been worked, *i. e.*, forged, hot or cold-rolled, drawn, or extruded.

2. These failures are independent of composition.

3. The fractures occur, in general, transversely to the direction of working.

4. Failures occur in brass materials which are under initial stress provided these stresses average over 5,000 to 6,000 pounds per square inch.

5. These stresses can be almost entirely removed by annealing at temperatures of about 300 degrees C. to 400 degrees C., which temperatures do not affect sensibly the ordinary physical properties.

6. No certain indication of the magnitude, or even the presence, of such stresses has been found in studies of the structure or of etching or corroding agents.

7. Brass may be manufactured practically free from stresses and still retain excellent mechanical properties, *i. e.*, will not be too soft.

Dr. Merica stated that investigations are being continued at the Bureau of Standards and invited the coöperation of both manufacturers and users of brass.

It will be noticed that neither in this present communication nor in the former paper has any reference been made to corrosion in general. This is, of course, a very important matter, and, so far as condenser tubes are concerned, is the subject of research by a Committee of the Institute of Metals, assisted by a Government grant. Without wishing to detract from this very useful work, it is thought that the subject of the physical properties of Copper Alloys, especially as regards their reliability and their adaptability for use for important structural purposes, is quite as worthy of a thorough research as is that of corrosion.—"The Steamship."

EMERGENCY REPAIRS TO A BATTLESHIP.

By Lieutenant E. E. Wilson, U. S. Navy.

From U. S. Naval Institute, Annapolis, Md.

The details of the work involved in the substitution by the ship's force of the *Arkansas*, of an electric motor for the wrecked starboard main circulating pump, have been here assembled for the reason that, while they demonstrate nothing new, at the same time they involve certain principles of electrical and mechanical engineering in a most instructive manner. In addition this work has emphasized one fundamental principle, namely, that all apparatus delivered to the service should be as near the commercial standard as possible, and that service operation should conform to the commercial standard, so that in emergency the resources of the whole country, rather than a limited reserve, are available for use. This point was demonstrated in the attempt to obtain a suitable motor from shore. Whereas most commercial motors are built for a voltage of 220–230, the Navy power circuits carry 125 volts, so that it was impossible to obtain on short notice a motor fulfilling the requirements. Instead it became necessary to adapt one of the ship's motors to the service, and take extraordinary precautions to maintain it in operation.

The main circulating pumps of the *Arkansas* are of the centrifugal type, and are driven by impulse turbines manufactured by the Terry Steam Turbine Co. of Hartford, Conn. The turbines developed about 75 I.H.P. at 600 r.p.m. when driving the pumps. Steam is taken from the auxiliary steam line at boiler pressure and expanded in the nozzles down to the back pressure of the auxiliary exhaust, about 15 pounds gage. Consequently the casing is not ordinarily subject to greater pressure than this back pressure and is designed on this basis. A relief valve, 3½ inches in diameter, is fitted to the exhaust side of the casing and set to lift at 20 pounds gage. At the time of the accident the pumps were being secured. The machinist had closed the throttle by hand and, upon seeing the pump stop, had closed the exhaust. Seeing steam leaking from around the glands, he started to close the throttle with a wrench, but before he could do so, the turbine exploded, blowing out the exhaust side of the casing as shown in the photograph. The relief valve had been frequently lifted by steam before the accident and afterwards relieved at a pressure of 20 pounds gage.

Examination of the turbine showed it to be damaged beyond repair by the ship's force, though the casing itself was the only part rendered unfit for use. Steps were immediately taken to rig some sort of substitute, and, in the meantime, the two condensers were cross-connected by way of the main drain so that the one circulating pump remaining could serve both condensers. This method limited the ship's speed to about 10 knots, with circulating water at the temperature of that locality, about 74 degrees F., though this speed could have been increased a knot or two with the colder water of higher latitudes.

Two alternatives presented themselves, (1) to drive the pump by chain or gearing from the main turbine shaft adjacent, and (2) to remove the old turbine and install a motor in its place. Of these (2) seemed the simplest and most reliable, provided a suitable motor could be had. Inquiry at the navy yard at Norfolk developed the fact that while no motor at 125 volts was available, there were several at 220 volts which might have been used. In order to use these, however, it would have been necessary to alter the armature circuits of one main generator, connecting the two windings in series rather than in parallel as they ordinarily are, so that the terminal voltage would be 250 instead of 125

volts. This method of operation was not an attractive one for the reasons that one generator would be required for the service of the pump alone, and that a separate circuit would be necessary from the dynamo rooms to the motor.

Since no suitable motor was available from an outside source, it was decided to take one from the ship itself. Investigation developed that the largest motor available was rated at 35 H.P., or about half that of the turbine. There were three types of motors of this capacity: (1) a shunt motor built for continuous service at 900 r.p.m., (2) a series motor

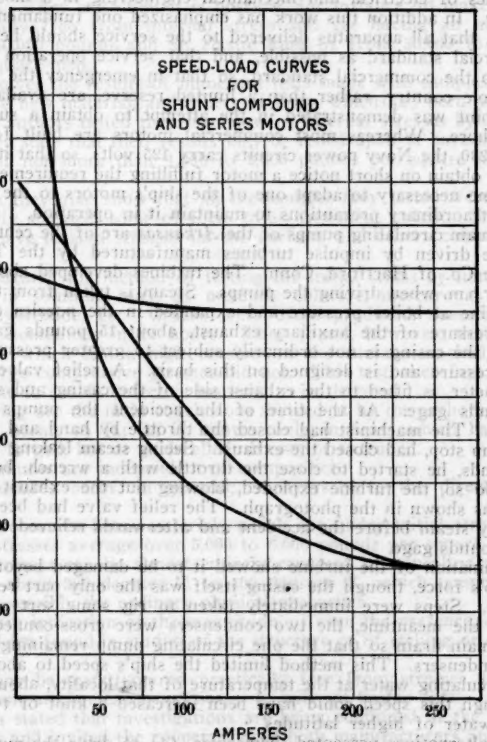


FIG. 3.

built for intermittent service at 350 r.p.m. and a compound motor for intermittent service at 350 r.p.m. The first belonged to a forced-draft blower, and was eliminated for the reason that 350 revolutions of the pump required the full power of the motor, so that it would draw an excessive armature current in its attempt to maintain rated speed. The series motor might have served, but it belonged on a boat crane and was very necessary to the ship for this service. The compound motor belonged to a deck winch aft, where it was used seldom and could therefore be

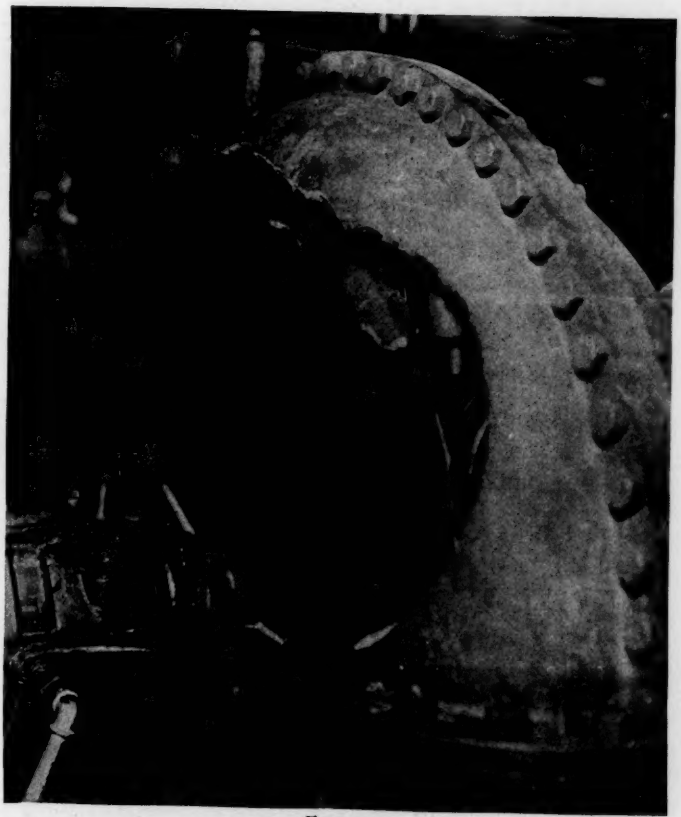


FIG. 1.

FIG. 2.

FIG. 4.

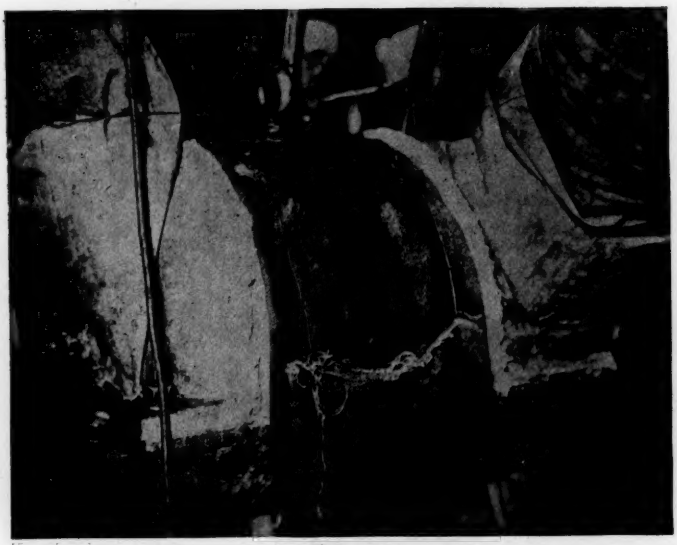


FIG. 5.

dispensed with. In addition its speed-load curve was favorable, though the limits of speed variation were necessarily restricted. It was also easily accessible and was therefore chosen for the duty.

The electrical characteristics that entered into the choice are shown in Fig. 3. Here are plotted (not to scale) the curves of speed versus armature current of the different types of motor. It was estimated that about 350 r.p.m. of the pump would require 35 H.P. output. At 125 volts this would call for an input of 245 ampères with an assumed motor efficiency of about 85 per cent. From the curves it is seen that the shunt motor was eliminated right here because at 245 ampères it would ordinarily run at 900 r.p.m. However, when the pump was driven at 350 r.p.m. it would require the full motor horsepower. As a result of the low speed the counter E. M. F. would be so small as to permit an excessive armature current to flow, burning out the motor, or blowing the safety devices. In addition to this it is evident that even if the motor had been able to carry this current, the shafting would have been over stressed, because of the greater torque required to transmit the same power at the lower speed. As for the other two it is seen from the curves that both the series and compound motors were adapted, as their rated speeds and armature currents corresponded to the speed and H.P. of the pump. The disadvantage of these two motors, however, lay in the fact that they were designed to carry full-load current for intermittent service only, while in the new duty they must operate continuously. However, since they were fully enclosed on deck it seemed reasonable to suppose that if opened up and properly ventilated below so as to get the heat away, they would be able to stand the load without burning. On this assumption, the deck-winch motor was removed with its controller, thoroughly overhauled and started below.

In the meantime the turbine half of the flexible coupling for the pump was bushed and fitted to the armature shaft so as to render the coupling up easy. At the same time the carpenter's gang had begun the installation of a wooden foundation of 6 × 6 timbers on the old foundation of the circulating pump which had been removed. This foundation was set up as nearly in place as could be from measurements of the motor, and certain holes in the planking were slotted so as to permit of adjustment fore and aft and athwartships after the motor was located in place. Adjustment vertically was had by leaving the top timber a little low and shimming up with strips of sheet iron. This foundation seemed amply solid as the vibration from both motor and pump was negligible. Since the pump itself had two rotors on the same shaft taking water from both ends, the end thrust was balanced and there was no need of a thrust bearing or of any stiffness to take up thrust. The long bolts were made in the shop while the motor was being sent below, so that by the time the motor was ready its foundation was in place.

The repairs to the motor consisted of the renewal of a shunt field coil, both bronze bearings and the armature, together with necessary varnishing and cleaning. The object of this work was to give the motor every chance to stand up under the work it was to do, and the necessity for these precautions became very evident when the motor was put in operation. At the same time it was much handier to send it below in sections as the distance from its old location to the new one was quite long. While the motor was being assembled in place the electricians were installing the starting rheostat and connecting the leads to the circuit for number six turret which passed through the wiring passage close by. A voltmeter and ammeter were installed in a convenient place, so that by the time the machinists had made the alignment everything was ready for operation.

In aligning the motor and pump use was made of the large flexible coupling. By means of the slotted bolt holes the motor was moved about until the flanges on the two shafts were parallel and their edges in line. After turning the motor over free to check the alignment, the coupling was bolted together and the motor started up. Predictions as to speed and load were immediately exceeded, as the pump was driven at 400 r.p.m. with 250 ampères at 120 volts, but the motor promptly began to heat. Inasmuch as the armature current was quite large it was assumed that the heating was due to this, but it soon became apparent that the trouble was in the bearings. Investigation showed that the new armature shaft had been made with the collars, which limited the end play, in such a position that there was no end play of the armature permitted. This was overcome by putting a gasket between the shields which held the bronze bearings and the casing so that the end play became about ¼ of an inch. This relieved the difficulty when running free, but as soon as the pump was coupled up the bearing nearest the coupling ran warm again and the whole motor became quite hot. It was then observed that in drawing up the coupling the shoulder on the shaft was drawn hard against the bearing and that this was the source of heat. The coupling was then broken and the motor shifted toward the pump until, when the coupling was drawn tight, the armature end play was halved between bearings. As a result of this, the armature, while running with full load, worked fore and aft under the flexible coupling, so that it was not only free in its bearings but permitted a certain variation in the brush position on the commutator which was of value in keeping the latter cool. This eliminated all bearing trouble.

In the meantime every effort was being made to keep the motor cooled off. The commutator end of the armature was cooled by a blast of air from the ventilating system, which was led in by a canvas chute. The forward bearing was under a blast of the same sort. At both ends the blast as first installed blew directly across; but when this failed, holes were drilled in the forward-bearing housing, and the air which went in at the commutator end passed the length of the armature and out at the forward bearing where it met the second blast and was carried away. This, of course, made the air about the bearing very warm at the same time it was rubbing on the shoulder of the shaft. When, therefore, the adjustment was made for halving the end play, the process was reversed. The chute on the forward end was secured to the shield so that all its air was forced through the holes toward the commutator, where the blast from the second chute, sweeping across, carried it away. This was a fairly satisfactory arrangement, but even with this, though the bearings remained cool, the heat was not getting away from the field coils and armature. A third air chute was then made to deliver a blast directly across the casing at its center, and while this helped on one side the other was still warm.

Finally it was decided to resort to water cooling, and this was accomplished in a very simple way. First, all possible leaks to the casing were plugged, then a ridge of putty was placed around the casing at each end. Next the intervening space was wrapped with a few turns of burlap and a small pipe was led from a handy salt-water pipe with a valve directly over the center of the motor. From this source the burlap bagging was saturated, and as the air blast swept across the motor, the water was quickly evaporated. The heat necessary to cause the evaporation was taken from the casing itself, so that a rapid flow of heat from the coils and armature was established. Of course, the latent heat of evaporation being large, a much smaller amount of water was required than if the water simply ran over the casing and increased its sensible heat. It was

possible, if desired, to so regulate the rate of flow to the casing that it was equal to the rate of evaporation, and thus prevent water from dripping to the deck. At any rate, the motor promptly cooled down and remained cool thereafter. Thermometers had been installed, one in each bearing, as shown in the pictures, and one in the path of the air discharged across the commutator. While these were not located to give accurate absolute results, still they were a good indication of the conditions at each place and thus were guides for operation.

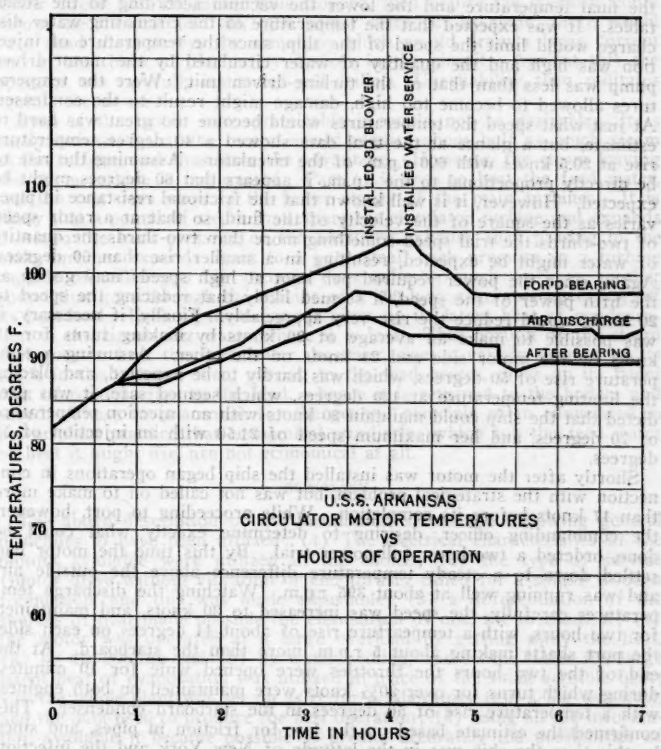


FIG. 6.

When the installation was completed and found running well, an electrician's watch was established on the motor and a test begun. From the curves, Fig. 6, it is seen that the motor gradually increased its temperature along a typical warming curve; then settled down and maintained this temperature. The final temperature was that at which the difference between the motor and the outside air was such that the heat was radiated as fast as it was generated and equilibrium was maintained. The actual temperatures then were dependent upon those of the air supply, and it was found that the curve of temperatures followed that of

the outside air while cruising. When the ship moved to northern waters, then, motor temperatures went down promptly, and no more trouble was had from this source.

The change of latitude, of course, had the same effect on the vacuum. The motor, operating at 400 r.p.m., was only 100 turns slower than the turbine on the other side. For a given speed the vacuum depends upon the temperature of injection, since the same quantity of steam condensed causes the same temperature rise, and the higher the injection the higher the final temperature and the lower the vacuum according to the steam tables. It was expected that the temperature of the circulating-water discharge would limit the speed of the ship, since the temperature of injection was high and the quantity of water circulated by the motor-driven pump was less than that of the turbine-driven unit. Were the temperatures allowed to become too high, damage might result to the condenser. At just what speed the temperatures would become too great was hard to estimate, but a glance at the trial data showed a 40 degree temperature rise at 20.5 knots with 600 r.p.m. of the circulator. Assuming the rise to be directly proportional to the r.p.m., it appears that 60 degrees might be expected. However, it is well known that the frictional resistance in pipes varies as the square of the velocity of the fluid, so that at a rotor speed of two-thirds the trial speed something more than two-thirds the quantity of water might be expected, resulting in a smaller rise than 60 degrees. Again, since the power required per knot at high speeds may go up as the fifth power of the speed, it seemed likely that reducing the speed to 20 knots would reduce the rise very appreciably. Finally, if necessary, it was possible to make an average of 20 knots by making turns for 19 knots on the motor side and 21 knots on the other. Assuming a temperature rise of 50 degrees, which was hardly to be expected, and placing the limiting temperature at 120 degrees, which seemed safe, it was predicted that the ship could maintain 20 knots with an injection temperature of 70 degrees, and her maximum speed of 21.50 with an injection of 55 degrees.

Shortly after the motor was installed the ship began operations in connection with the strategical problem, but was not called on to make more than 17 knots before its completion. While proceeding to port, however, the commanding officer, desiring to determine exactly what could be done, ordered a two-hour, full-power trial. By this time the motor had settled down to a steady temperature difference above the outside air, and was running well at about 395 r.p.m. Watching the discharge temperatures carefully, the speed was increased to 20 knots, and maintained for two hours, with a tempearture rise of about 44 degrees on each side, the port shafts making about 5 r.p.m. more than the starboard. At the end of the two hours the throttles were opened wide for 10 minutes, during which turns for over 20½ knots were maintained on both engines, with a temperature rise of 50 degrees in the starboard condenser. This confirmed the estimate based on the law for friction in pipes, and since by this time the ship was in the latitude of New York and the injection was 54 degrees, the final temperature was but 104, leaving a wide margin had it been necessary to work up to the maximum speed. Upon arrival in port the motor was shut down after over five days' continuous operation, being in condition to continue indefinitely.

The motor having proved successful, there were consequently many people who advocated the installation of motors for this service in place of the old turbines. Against this proposition there were two important arguments. In the first place, the engines became dependent upon the operation of a generator at a distance from them, over which the engine-room force had no control. The blowing of a circuit breaker or an

accident to the generator would stop the flow of water through the condenser, and such an accident while operating at high powers might result in burning up the condenser before the engines could be stopped. To avoid this two generators were kept in operation and a careful watch was stood at all times. A signal was arranged between the pump and throttle, so that if the motor stopped for any reason, the officer of the watch would be warned in time to shut down the engines and save the condenser. If through design or accident the power were shut off the motor circuit and the motor thus stopped, then before the operator could throw the rheostat to "off," the power were cut in again, there would be a dead short across the armature which would draw a large current and at best blow all the safety devices in the circuit. Against this the operator was carefully instructed and drilled. Unreliability, then, militates against the motor.

Even if the motor were practicable it would not be desirable. It is easily estimated that the motor would require about half as much steam as the turbine, and thus have the advantage. However, the steam leaving the pump turbine is not wasted. Part of it goes to heat the feed water, and the rest does work in the lower stages of the main turbines. It is likely that a pound of steam that enters the circulator turbine and expands down to the back pressure of the auxiliary exhaust, then passing to the main turbines expands again to a very small back pressure in a turbine capable of utilizing a big vacuum, does quite as much work as another pound that goes through the turbine of a 300 k.w. generator operating off its best load. Each of these pounds gives up its latent heat of condensation to the circulating water where it is lost. That portion of the steam that goes to heat the feed water, however, does a certain portion of its work in the turbine; then, passing to the feed heater, gives up its latent heat of condensation to the feed, whence it is returned to the boilers and saved. These facts account for the general principle often enunciated in power-plant practice that economical auxiliaries which rob the feed water of heat it might use, are not economical at all.

SUMMARY.

The above description of the installation shows the following general principles to have been utilized: (1) The electrical characteristics of the motors available were utilized in the choice; (2) the type of service (motor drive without vibration or end thrust) called for a simple wooden foundation, and this was made adjustable; (3) a differentiation of the sources of heat made possible the elimination of one, the mechanical; (4) the motor was operated at a large overload by getting away the heat, (a) by air cooling and (b) by water cooling, and in the latter the large latent heat of evaporation was utilized in place of the small sensible heat; (5) the whole plant was set up as if for a laboratory test, a log was kept, and the data used to produce the best operation; (6) using all the information available it became possible to predict accurately the possibilities of the plant, then to confirm the prediction by actual results.

CONCLUSIONS.

In general two conclusions can be drawn from the case: (1) an impulse turbine, since it is not designed to withstand all the stresses to which it may be subjected either by design or accident, must be well protected by safety devices, and the same care must be given these devices as is bestowed upon boiler safety valves, including routine lifting by hand and steam; (2) were the wiring of ships installed similar to that ashore, where by means of a three-wire system 110 volts can be had across two

wires for lighting and 220 volts across two others for power, and were the appliances made to conform closely to a commercial standard, then the resources of the whole country would form a reserve from which to draw equipment in times of stress.

THE SIMPLEX PATENT SALT DETECTOR.

The presence of salt in boilers is now universally admitted to be highly objectionable because of the reduced steaming capacity which results from the deposit of salt on the heating surfaces, and also because of the danger of overheating which is always present when there is scale on the surfaces. In the case of water-tube boilers where even a small deposit of salt may have serious consequences, greater precautions are generally taken to keep the feed fresh than with the ordinary type of boiler.

In the ordinary course, the water in the boiler is tested regularly by the salinometer. These tests are satisfactory so far as they go, but give no indication of the rate at which salt is entering the boiler. It is quite evident that the salinometer test may give the same result when there is a very small leakage of salt water into the feed supply for many hours, as it would be if there were a very large flow of salt water for a short time, as, for example, would occur in the case of a split tube in the condenser.

The Simplex Patent Salt Detector is designed to give a *continuous* indication of any salt in the feed water, and thus show the presence of salt *before* it enters the boiler; also in the event of a sudden increase of salt—as may occur owing to a split tube, or the evaporator priming—it gives warning immediately, and so allows the defect to be remedied before the density of the water in the boiler increases perceptibly. The apparatus is a very simple one, and its working is based on the well-known fact that pure water has a high electrical resistance, which, however, falls very quickly if certain compounds—of which ordinary salt is one—are added to the water. The essential part of the apparatus is a glass vessel fitted with two electrodes by which a current is passed through a sample of the feed water. The necessary current is taken from any convenient power or lighting circuit, either direct or alternating current. The electrodes are made of a special grade of carbon and are so arranged that no metal part of the connections is in contact with the water. This is essential to avoid the excessive corrosion which would otherwise take place. The electrodes are adjustable so that the sensitiveness of the instrument can be varied. The instrument gives a continuous indication of the saltness of the feed, and this is done by having a supply of feed water flowing through the vessel—entering at the bottom at one end and flowing out at the opposite end.

The resistance of the water falls very rapidly for a small increase in the amount of salt present, and this tends to make the simplest arrangement too sensitive. To obviate this, in the Simplex apparatus a resistance is introduced which is preferably in the form of an incandescent lamp. By varying this resistance, or in other words by using lamps of different candle power, the sensitiveness can be varied as desired. A high candle-power lamp gives greater sensitiveness, and a low candle-power less sensitiveness. The voltage of the lamp should be slightly lower than the line voltage—about 5 to 10 per cent is a good average value. The lamp shows no light when the water is fresh, but as soon as salt is present will begin to glow a dull red, which gradually increases to full brightness as the salt increases. The brightness of the lamp enables the officer on watch to see at a glance whether there is salt in the feed and also whether it is in sufficient quantity to require immediate attention.

The voltmeter gives a quantitative indication of the amount of salt present, and may be calibrated, if desired. This cannot be done, as a rule, with any great accuracy, however, in the majority of cases, as the readings vary considerably with the temperature of the feed water. In many cases it is desirable to have a definite warning given immediately the salt exceeds a fixed percentage. This is given by a bell or buzzer, which is arranged to be operated by a relay connected in series with the instrument. The relay is very easily adjusted so as to operate at the desired density.

The whole apparatus is self-contained in a polished teak case with a plate-glass door. The buzzer is fitted to the outside of the case, while the connection to the electric circuit is made by a plug, so that it is only necessary to insert the plug in the socket, and take a connection from any lamp holder. Should a log be required showing the condition of the feed throughout a watch, or any other period, this can be arranged by having a recording voltmeter with a roll or circular chart operated by clockwork in the usual way.

The sensitiveness of the instrument is very great, and compares well with the nitrate-of-silver test. One grain of salt per gallon can readily be shown. It is thus very suitable for use in connection with the distilling of fresh water for drinking purposes.

A series of tests have recently been carried out at the National Physical Laboratory at Teddington, for different distances apart of electrodes and with different degrees of sensitiveness obtained by using lamps of different candlepower in circuit. The curves thus obtained show clearly the sensitiveness of the instrument.—"Shipbuilding and Shipping Record."

REDUCING SPEEDS MAGNETICALLY.

A NEW STEP-DOWN GEAR ADAPTED FOR USE ON TURBINE-DRIVEN SHIPS.

A very ingenious gear has recently been invented for use in turbine-driven ships to permit a high-speed turbine to drive the propeller shaft at its most efficient speed. This gear is magnetic, and it operates on an entirely new principle. It can be designed for a great range of reduction ratios, the apparent limits being about 6 to 1 and 40 to 1, and the higher limit can be obtained in a single step.

The principle of the gear can best be understood by referring to the diagram, Fig. 1, which represents a typical design. Here we have the driver or primary field element *A*, which is rotated by suitable clutch connection with the turbine shaft. This field element, which consists of two poles, is energized by some suitable source of current. Surrounding it are two stationary cages or stators, *B* and *C*, the stator *B* consisting of magnetic bars of laminated sheet iron mounted on non-magnetic supporting rods, while the stator *C* is built up of toothed sheets of iron. The rods and teeth are magnetically energized as they come within the field of the primary element, so that as the latter rotates it sets up a wave of magnetism that sweeps around the stators. Between the stators is the driven member *D*, also made up of bars of laminated iron. It will be observed that there are 64 magnetic bars or teeth on each stator, while the secondary or driven member has 66, so that at only two points are the teeth of the stators and the secondary in alignment. With the parts at rest and under no load, this alignment will be formed in the field of greatest magnetic flux—in other words, directly in line with the poles of the driving member, as shown at *E* in the diagram.

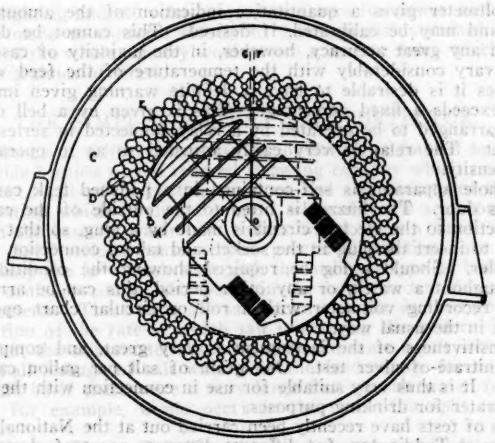


FIG. 1.—DIAGRAM ILLUSTRATING THE OPERATION OF THE MAGNETIC REDUCTION GEAR.

Now, let us suppose that the driven member is held stationary while the driving member is rotated to the position *F*, as shown in broken lines. As at this point the teeth of the stator and the secondary will be out of alignment, the magnetic force will set up a torque between the teeth which will only be satisfied when the secondary is permitted to rotate far enough to bring its teeth into coincidence with the teeth of the stator and at the point of greatest magnetic flux. In other words, while the driver is moving through the angle *E O F* the driven member moves through an angle *G O F*. In a complete rotation of the driver there will be a movement of the secondary through the space of two teeth; in other words, there will be a speed reduction of 33 to 1. Of course, the driven member will lag behind the driver by an amount depending upon the load it has to carry.

From a brief study of this diagram it will be noted that the reduction ratio is equal to the number of secondary bars divided by the number of poles in the primary, while the number of stator teeth in each cage must be smaller than the number of secondary teeth by the number of poles of the primary. On a moment's consideration, it will be evident that if the number of teeth in the secondary is greater than that in the stator, the secondary will rotate against instead of with the driver. The torque on the secondary is not exerted by the primary, as in a magnetic slipping clutch, but is obtained by the pull of stator teeth on the secondary bars, while the primary merely furnishes the magnetic force. The secondary and stationary teeth form as many alignments as there are poles, so that if the driver is provided with more than two poles the ratio between the secondary and the stator teeth must be such as to provide the same number of alignments. Then, when the field poles are magnetized, the secondary will assume a position of greatest magnetic reluctance—that is, they will move into alignment with the stator teeth where the magnetic flux is greatest. Thereafter the field cannot move without a corresponding movement of the secondary at an angular velocity determined by the reduction ratio.

This new magnetic gear is characteristic of a mechanical gear in that it has a fixed ratio of reduction for which it has been designed, and in this respect it is like a synchronous motor which when supplied with an alternating current of fixed frequency will rotate at a corresponding speed; in the gear it is the speed of the driver, in place of the frequency, which determines the speed of the secondary or driven element. It resembles the synchronous motor also in its ability to carry a certain maximum load before falling out of step with the driver. In designing, therefore, a suitable overload margin must be allowed above the normal operating load.

In order to corroborate the working of the principle of this gear an experimental machine was built, and to save time a stator and other parts that were on hand were used in its construction. A photograph of this machine is reproduced herewith. It has a field of six poles. The stator

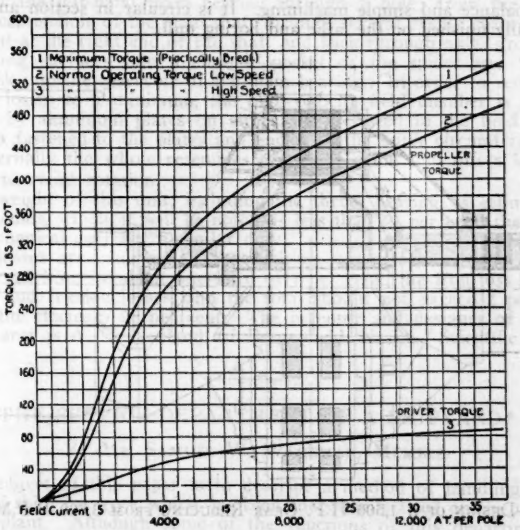


FIG. 2.—CHART SHOWING THE OPERATION OF THE EXPERIMENTAL MACHINE.

consists of a single cage, which surrounds the secondary. The inner stator, indicated at *B* in Fig. 1, was omitted from this machine. There are 33 bars or teeth in the secondary and 27 on the stator—the reduction ratio, therefore, is 5.5 to 1. The operation of this gear is indicated in the accompanying chart, Fig. 2, in which the curve *1* represents the maximum loads the machine will carry with corresponding ampère turns per pole. It will be noted that the first part of the curve resembles the torque characteristic of a series motor, while the latter part is much like a saturation curve and indicates that by increasing the pole and stator sections, the increased force beyond the bend of the curve would be made available for increasing the flux density at the bar gap and therefore adding materially to the torque of the secondary.

Based on the tests of the experimental machine, designs have been prepared for a full-sized unit, which is shown partly in section in Fig. 3, while Fig. 1 represents the arrangement of poles and teeth in the primary, secondary and stators. In the two drawings the same reference numerals are used for corresponding parts. This gear is arranged for a normal horsepower of 1,500 and an overload margin of normal field of 60 per cent., reducing the speed from 3,600 r.p.m. to 109, a reduction ratio of 33 to 1. This machine contains an additional stationary cage inserted between the field and secondary, the purpose of which is to increase the torque on the secondary. A torque practically double in value is produced on the secondary without increasing the core length. However, the field ampère turns must be increased to supply the necessary force for the added gap. A substantial cut in weight and an increased efficiency are the result.

The field has been especially developed to fill the requirements of high speed, balance and simple machining. It is circular in section and can be practically finished on the lathe and boring mill.

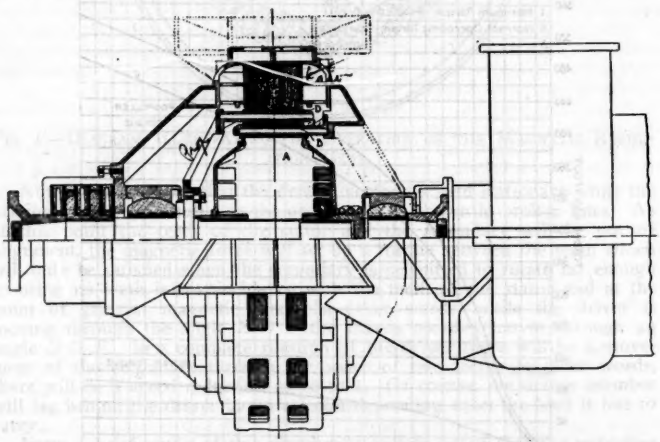


FIG. 3.—DESIGN OF A 1,500-H.P. GEAR REDUCING FROM 3,600 R.P.M TO 109.

The field copper, wound directly in circular grooves turned in the core, comprises a bare strip insulated from the core by substantial insulating plates and between turns by a strip of fish paper. The insulating plates are of such thickness as to wedge the strip in the groove under tension, so that once wound and held by a bronze wire hand, the considerable centrifugal forces will be unable to displace it and thus cause unbalancing. The separating flanges serve to form a plurality of grooves to prevent a cumulative strain on the outer flange and to make possible a strip of reasonable width. Each groove contains two layers separated by an insulating plate and wound in opposite directions so as to bring all leads to the outside for easy connection.

Particular attention has been paid to the design of the cages. Their present arrangement combines the greatest possible section in the laminated bars with maximum strength of the non-magnetic supporting rods. A non-magnetic ring having axial holes drilled through, into which the rods are pressed, serves as the supporting member for the laminated bars,

each of which is forced between adjacent rods and insulated from them by a fiber tube on the rods 1/16 inch in thickness.

The rings, in turn, are screwed to the spider of the secondary and the stationary end bracket of the stator respectively. A narrow ring, insulated from the rods, ties each of the cages on the other side and thus makes the cages thoroughly rigid. Screwed to the spider of the secondary cage is the main or driven shaft, which extends through the bore in the field so that the field is partly supported by and rotates around the main shaft on bearings provided on either end. The two main bearings, one at the end of the main shaft and the other on a stub extension of the field sleeve, support the weight of the primary and secondary and thereby make the gear unit independent of the turbine and propeller-shaft bearings.

The main, and particularly the internal field bearings, are oiled by pressure circulation, the oil being introduced into a circular groove in the center of the left-hand main bearing through a hole; from the groove into the center hole of the main shaft through several evenly spaced radial holes; out at the right end of the shaft and back through axial grooves in the bearing surfaces into a circular pocket of the spider and through holes back into the reservoir of the main bearing. In order to reduce the windage losses to a minimum, the high-velocity field member is entirely enclosed by aluminum plates on either side screwed to the field with a thin strip fastened to the plates and enclosing the outer circumference, so that externally the whole resembles a smooth drum and offers little resistance to rapid rotation.

The weight of this unit, including the thrust bearing, is estimated at 24,000 pounds, which, it will be noted, is less than 50 per cent. the weight of a mechanical gear for a similar duty.

The losses are made up of the exciting loss, which in this design is normally 4.5 k.w., or about one-third of 1 per cent., the iron loss and the windage and friction losses, and the total should not exceed 2 per cent., making the efficiency 98 per cent. The inventor and designer of this ingenious gear is A. H. Neuland, of Bergenfield, N. J.—"Scientific American."

THE SERVICE DEPARTMENT OF A SHIPYARD.*

By Mr. Samuel W. Wakeman, Member.

The object of this paper is to describe a method of handling certain phases of shipyard work which has been successfully tried out in one large eastern plant. Although some of the functions of this department are probably being carried out in the different yards today, we do not know of anyone who has treated this problem in just this manner. About two years ago the president of the plant became interested in the enormous number of men that it was necessary to hire each month in order to maintain the working force then needed to carry out the work in hand. He determined that this problem must be solved, and a committee was selected from the plant, composed of workmen, foremen and others. They made a tour of the country, visiting the following manufacturing concerns: Link Belt Company, Sherwin Williams Company, Jeffery Manufacturing Company, Cleveland Hardware Company, Curtis Publishing Company, Chalmers Motor Company, Cleveland Foundry Company, Goodyear Tire and Rubber Company, Cadillac Motor Company, National Cash Register Company, Dodge Motor Company, Packard Motor Company, Ford Motor Company, Westinghouse Manufacturing Company, Car-

* Paper read before the Society of Naval Architects and Marine Engineers, in New York, November, 15.

negie Steel Company. The most courteous treatment was accorded this committee at each of the above mentioned plants, and the fullest information was given by them on all of the points which this committee investigated.

On their return they prepared a report of their investigations, together with such recommendations as to the betterment of the then existing conditions as they thought should be made, these recommendations including such ideas as the centralizing of employment instead of allowing each one of the different department heads to carry this on themselves. It also recommended that a new building be erected for this purpose; that a different point of view be taken with regard to applicants for positions, namely, their treatment should be as courteous and as fair as was possible to be given; that the housing and boarding situation be investigated; that all men leaving the employ of the company should be interviewed and given a chance to air their grievances. They further recommended a great many lines of activities to be encouraged in the plant, and touched on many lines that had to do with the physical, mental and social condition of the employees. The recommendations of this committee were printed in pamphlet form and sent to about 200 people in the plant. They were asked to freely criticise and, if possible, to make further recommendations. All of this information was then collected and formed the basis for the policy which is being pursued at the present time in regard to the service work in this plant.

A brick building was erected near the main gates of the plant to take care of the hospital work, the employment department, and the general activities of that nature. These activities were grouped under the name of Service Department, which, as its name signifies, is to render service to the employees of the plant. This department was put under the supervision of the general superintendent of the plant. The scope of its activities are as follows: Employment, hospital, safety work, educational work, apprentices, employees' club, works paper, suggestion committee work, yard restaurant, shop training. The personnel of this department consists of the following: Supervisor, assistant, clerks, stenographer, supervisor of instructors, instructors, employment agent, employment clerk, doorman, doctors, nurses, messenger boys for conducting applicants to shops.

In order to see clearly the relationship between the Service Department and the superintendents and foremen in the yard in regard to the supply of labor, the following illustration is used: The ship carpenter may want 10 pieces of 12-inch by 12-inch yellow pine, 30 feet long. His requisition comes to the purchasing agent, who buys this material and has the same delivered to the ship carpenter. If on inspection by him three or four pieces out of the 10 prove defective, he condemns this material and the purchasing agent replaces it. The ship carpenter decides what class of work this lumber can be used for and decides also whether the price paid for it is justified.

The same procedure holds for the supply of labor. A foreman finds he will need a certain number of men at a given time. He sends his requisition for this labor to the employment division of the service department. They send the required number of men to the foreman after interviewing these men and weeding out, so far as is possible, such applicants as would be incapable of carrying out the work for which they are required. These men are sent to the foreman of the department, who takes or rejects whatever number he may see fit and rates those he keeps, and those he rejects are sent back to the employment department, where an effort is made to place them in some other department in the yard where they may find work which they can do.

If the men hired are not satisfactory to the foreman he may discharge them. In no particular are any of the foreman's prerogatives in regard to

Head of Service Department.

- Employment Agent — Interviewing and hiring new employees. / Recruiting labor outside the plant.
- Correspondence Clerk — Housing new employees. / Interviewing and recording departing employees.
- Messengers (3) — Monthly analysis of employees hired and departed.
- Recording Clerk — Recording on cards new employees. / Daily report of men hired.
- Door Man — Filing record cards of new employees. / Charge of main waiting room.

Assistant Stenographer, 2 Clerks.

- Employees, Magazine — Associate editors. / Collection, compilation, proof-reading, publication and distribution.
- Safety Work — Monthly statement of hours lost and percentage; accidents; recommendations approved; by General Safety Committee. / Investigating and reporting special cases, bulletin posting. / Shop committees, organization and reporting. / General Safety Committee minutes.
- Apprentices — Applications, tools, bonuses and certificates. / Shop credit reports. / General welfare, social activities, &c. / Book apprentices time and quarterly reports.
- Employees' Club — Minutes of meetings. / Publicity of activities. / Membership and dues. / Athletic activities.
- General — Circulation of monthly periodicals. / Restaurant committee, supervision of yard restaurant. / Arrangement bulletin boards. / Naturalization assistance. / Navy League, rifle company. / Assistance to employees in promoting outings. / Information of any kind to employees.
- Educational — Keep records of attendance. / Outline of curriculum. / Receive applicants for classes.
- Housing — Listing of lodging and rooming houses available for new men. / Information regarding houses for sale and to rent. / Advice to employees wishing to buy houses.
- Suggestion Committee — Receipt and referring of suggestions to parties directly concerned. / After receipt of comments, action upon same by committee. / Recommendation of an award. / Notification to sender.

the handling of labor taken from him. The advantage of this method, however, is that the foreman is not compelled to interview a large number of men who have no qualifications for the work he is supervising, and his time is conserved for the more important duties he has to perform.

Each applicant for a position is put through a careful physical examination by the company's doctor, and only such men are employed as are capable of doing the work for which they are hired. This has resulted in getting a very good class of men, and, where slight physical defects are found which can easily be remedied, the man is sent to the proper medical authority and taken care of. At first there was considerable objection on the part of some of the men, but at the present time practically no one objects to the physical examination.

The employment department keeps an up-to-date requisition of all labor requirements of the plant, and takes such steps as are necessary,

through its various sources of supply, to keep this requisition filled. It attempts to learn, by keeping in close touch with the various departments, the probable future needs for labor, which is a vitally essential factor in keeping a steady flow of men into the yard. All applicants for work are quartered in a waiting-room in the service building, equipped with chairs, and are interviewed in turn by the employment agent, being given a private interview in a separate room. Each applicant when hired is given a book which contains all the information about the plant that it is considered a new man should have. This book has been carefully prepared and gives the hours of labor, the method of tool delivery, certain safety suggestions, times and methods of payment of wages, and, in general, information in a condensed form which the new employee generally gets only after long association in the plant. This book has proven to be a great help to new men coming into the plant. All employees transferred or leaving the employ of the company are required to be interviewed by the employment agent before they can obtain their wages. This provides an opportunity to place the man in a department where he may fit to better advantage than the one in which he was working, and also gives him opportunity to state any grievances he may have in connection with the company, his foreman or his work. These grievances are carefully tabulated and looked into and have been found to be a great source of valuable information in connection with controlling the feeling the employees have toward the company. All records for present and past employees are filed alphabetically in the employment bureau, and these records contain the name, age, trade, physical qualifications, address, place of past employment, rate, etc. There is a daily report prepared by the employment department, showing the number of men hired and leaving by trades, also daily gain or loss in the working force. The department also makes a monthly report of the turnover of labor, including the number of men hired and leaving by trades, showing the length of service and reasons for the men leaving.

One of the most important phases of the employment work is to provide proper housing facilities, and to this end the employment department lists, for the information of all applicants, boarding houses and available houses for sale or for rent. If there is no position open at the time the applicant applies in person or by letter, an application card is filled out and filed by trades. All applications for employment are filed, and, as far as possible, the applicant is informed as to what date his services will be required, giving the minimum and maximum rate for the trade desired and the hours of labor.

Under the service department is a competent surgeon and an up-to-date hospital, where all of the accident cases in the yard are treated. The families of the employees may obtain free medical attention during certain hours of the day. The safety work in the plant, which had reached a fair state of development, was combined with the work of the service department and the different shop committees, together with the General Safety Committee, use this building as a meeting place, where frequent talks by men who are specialists along this line of work are given to these men. Each department has its own safety committee, which makes periodical inspections and reports directly to the General Safety Committee. A great many ideas for work of this nature have originated in these committees and numerous appliances and safeguards have been adopted, due to suggestions received from them. All accidents are carefully investigated and the causes are immediately remedied; responsibility for accidents, as far as possible, is placed and steps taken to rectify them.

All of the educational work in connection with the training of apprentices and the training of men in the shops and different departments throughout the yard have been placed under the jurisdiction of the service department. During the winter months, evening schools are conducted

which provide training in naval architecture, mould-loft work, plumbing, coppersmith work, machine shop design, mathematics and blue-print reading. The instructors for these evening classes are, for the most part, drawn from the various departments in the yard and the instructions carried out under the direction of the local school board. These classes are open to any male beyond the age of 16 in this locality, and are attended by mechanics, apprentices and helpers to the number of about 200.

All of the work in connection with the apprentices in the plant is carried out under the supervision of the foremen in the different departments, but along the lines as laid down by the educational part of the service department, and the head of this department has been very successful in this connection with the attitude of the apprentice toward the company. Each year a dinner, to which prominent officials of the company are invited, is given by the apprentices, and efforts are made to promote the best of feeling among the young men who are being trained for future service in the company.

The service department supervises the publication of a works paper, which is issued every three months by the employees. This paper has developed to such an extent that its issue is awaited with a great deal of interest by all of the men in the plant. The company presented to the employees of the plant a club house equipped with bowling alleys, rifle range, assembly and dancing hall, pool table, a band room and an athletic field, where the different athletic teams practice and have their games. Quarters are provided for the athletic teams of the company in this building, with suitable shower baths, etc. This club is run entirely by the employees of the company, and only such necessary supervision is given to it by the service department as is absolutely necesary. A restaurant is maintained inside the plant for the benefit of the employees. The supervision of this restaurant, although run by an outside caterer, is left to a committee of the employees, who are in close touch with the service department.

One very important part of this service department work is the handling of the suggestions sent in by employees. Boxes are placed throughout the yard, and prizes are offered for suggestions sent in by the men. There is a great deal of work in connection with the proper understanding of what the employee means, the making of sketches, the referring of these suggestions to the proper people so that they may be acted on, and the attitude which the men have toward this phase of the work must be carefully handled in order that the greatest amount of good may come from it. The work of this committee, of which the head of the service department is chairman, is one of the very important functions of this department.

All of this work has begun to show results which it is felt can directly be traced to the work of this department. Statistics for many of the large manufacturing plants throughout the country that have attempted to increase their working force have shown that it is necessary to hire anywhere from seven to ten men in order to obtain one permanent employee. The latest results here have shown that it has been necessary to hire two men instead of from seven to ten in order to obtain one. When you consider that it is estimated that it costs somewhere between $20 and $35 to hire a new employee, the result of cutting down the turnover in any plant means a tremendous saving in dollars and cents, although it may not always be possible to measure this directly. Much of the reason for this decrease in turnover can be credited to the selection on the part of the service department; the hearty coöperation of the foremen, who have allowed a very liberal system of transfer from one department to another; the training of men into trades which they were not familiar with, and, in the main, to the general broad policy of the man directing the company.

Before the above plan was in operation, in order to maintain a force of about 4,000 employees it was found necessary to hire, in the course of a

year, 8,000 men. During the last six months the present force has been increased from 4,000 to 8,000, and this has been done by hiring 9,000 men. While these results cannot all be attributed to the work of this department, nevertheless it is felt that, in a large measure, the work which has been accomplished by them is responsible for a large percentage of this showing.

It is not claimed that any new or radical ideas have been adopted in connection with the handling of this situation, but what has been done is to find out what successful concerns in other lines of work have done, and to combine them together in such a way that they might be adapted to the work of a shipyard. A man was employed who was thoroughly familiar with the sources of labor supply in the district in which this plant is located, and much credit is due him for the enthusiasm with which he approached this difficult problem. An outline diagram showing the relationship of the different functions of this department to each other is given on a preceding page.

THE BEARDMORE MARINE ENGINE.

TWO-CYCLE, HOT-BULB TYPE OIL ENGINE DEVELOPING 160 BRAKE HORSEPOWER AT 280 REVOLUTIONS PER MINUTE.

One of the outstanding developments of recent times in marine propulsion is undoubtedly the greatly extending sphere of application of the semi-Diesel or hot-bulb type of oil engine, together with the rapid increase in power developed per unit. Quite recently this type of engine was universally regarded as being suitable for the development of only relatively small powers, say 25 brake horsepower to 30 brake horsepower per cylinder, but recent modifications in design and the virtues of rigid simplicity have brought it greatly into favor even for powers up to 130 brake horsepower per cylinder.

The great majority of marine semi-Diesel engines have points of similarity in the system upon which they work, and practically all are of the two-stroke cycle, utilizing the enclosed crank chamber for the compression of scavenging air. The "Beardmore" engine, which we illustrate, has been designed essentially to meet the peculiar requirements of marine propulsion, and is the result of much experience with this type of engine installed in yachts, coasters, lighters and fishing vessels. It is particularly suited to consume fuels ranging from 0.8 to 0.9 specific gravity, but can be adjusted to use either slightly lighter or heavier oils. Working on the two-stroke cycle, it develops 160 brake horsepower while running at 280 revolutions per minute. This relatively slow speed of revolution is, of course, conducive to high propeller efficiency. As all bearing surfaces are liberal and materials carefully selected, an outstanding feature of the engine is its capacity for long and hard service. The engine is directly reversible by means of compressed air, and requires no disconnecting clutch between the engine and propeller, the latter advantage being one that appeals very strongly to marine engineers in general and oil-engine users in particular. Figs. 1 to 3 give sections and elevations of the engine, Figs. 5 to 9 are details of the main valves, and Figs 10 and 11 are diagrams from the cylinder and crank chamber.

CYCLE OF OPERATIONS

In the main engine there are four cylinders, each 11 inches diameter by 15 inches stroke, and the cycle of operations may be briefly described as follows, the reference numbers being those shown on the section of one of the cylinders in Fig. 1: During the upward stroke of the piston, air is drawn through non-return valves into the crank chamber 1, to be com-

pressed on the downward stroke and admitted into the cylinder at the correct moment through scavenging air ports located directly opposite to the exhaust ports 2. These ports are covered and uncovered by the piston, on the top of which a deflector is arranged, to give to the air blast the direction most suitable for efficient scavenging. While the under side of the piston is performing the functions of a scavenging air pump, the upper side compresses the contents of the previously scavenged cylinder on the upward stroke in preparation for the downward or power stroke. A separate fuel pump is provided for each cylinder, and once in each revolution of the engine fuel is sprayed directly into the combustion chamber 3, through pulverizing nozzles fitted with automatic check valves, and ignition is obtained through the fuel impinging on the heated surface of the combustion chamber. The combined heat of compression and combustion sustains the temperature necessary for ignition, and consequently no blow lamps are required except for a few minutes before starting the engine. The various operations of the working cycle can be clearly understood from an inspection of the indicator diagrams taken from the cylinder and crank chamber and reproduced in Figs. 10 and 11, and the sequence and correlation of the operations will be appreciated. These diagrams show the excellent results obtainable by careful design based on extensive experience.

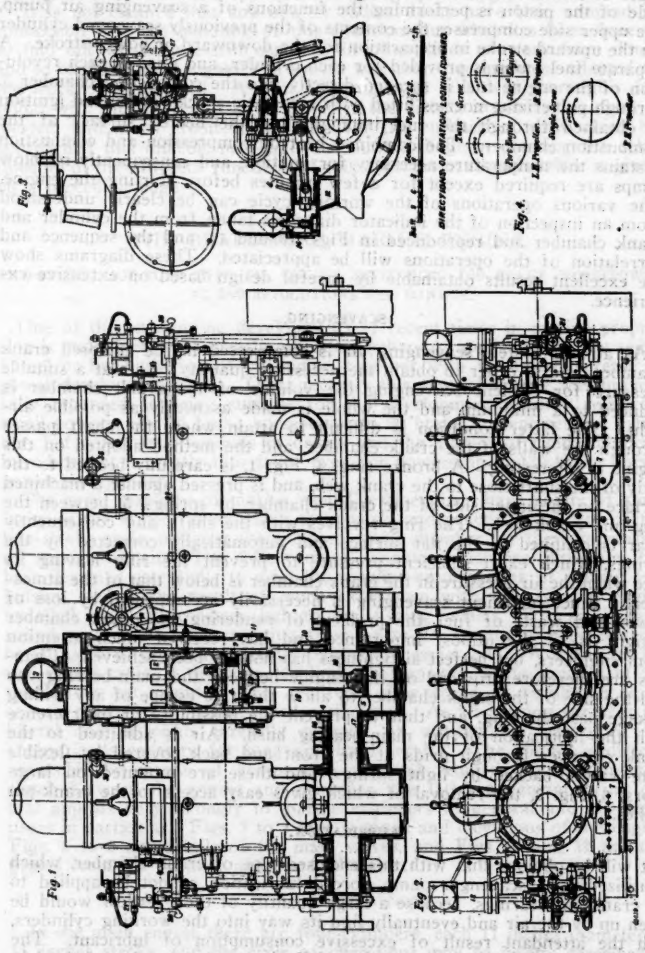
SCAVENGING.

As already stated, scavenging air is compressed in the enclosed crank chamber, and in order to obtain the necessary quantity of air at a suitable pressure for efficient scavenging the volume of the crank chamber is reduced to a minimum and the whole is made as nearly as possible airtight. The latter condition is difficult to attain where the shaft passes through the walls of the crank chamber, and the method adopted on this engine is interesting. A bronze ring, 4, Fig. 1, is carefully bedded to the body of the shaft close to the crank web, and is pressed against a machined surface on the inner side of the crank chamber by springs, 5, between the ring and crank web. The ring revolves with the shaft, and consequently wear is confined to the flat surface and automatically corrected by the springs, which exert sufficient pressure to prevent the ring leaving its face when the air pressure in the crank chamber is below that of the atmosphere. Since inefficient scavenging is necessarily accompanied by loss of power and waste of fuel, the problem of rendering the crank chamber airtight is of the utmost importance and has received much attention from engineers, but perfect airtightness has not yet been achieved. Channels are therefore provided on this engine between the main-bearing cap and the side of the crank chamber to allow the free escape of any trifling leakage past the ring, and thus to prevent the possibility of interference with the lubrication of the main-bearing bush. Air is admitted to the crank chamber through grids at the front and back covered by flexible plate valves backed by light springs, and these are mounted on large doors, 6, Fig. 3, the removal of which gives easy access to the crank-pin bushes.

LUBRICATION.

It will be obvious that with this enclosed type of crank chamber, which is utilized as a scavenger pump, forced lubrication cannot be applied to the crank-pin bearings, because a large quantity of the overflow would be taken up by the air and eventually find its way into the working cylinders, with the attendant result of excessive consumption of lubricant. The question of lubrication, therefore, is a difficult one, but has been solved in the "Beardmore" engine in a most satisfactory manner. A forced sight-feed system, with separate plunger pump and oil tube to each point where

lubrication must be positive and regular, has been adopted, and by this means the quantity of lubricant supplied to any point is definitely controlled by the attendant, while any interruption of the supply due to a choked tube or faulty valve can be instantly detected and the sight glass shows clearly which point is affected. Two eight-feed lubricator boxes, 7, Figs.



SECTIONS AND ELEVATIONS OF BEARDMORE MARINE SEMI-DIESEL ENGINE.

1 to 3, are employed on this four-cylinder engine, and the reciprocating drive is obtained from the fuel-pump actuating gear. The sixteen feeds are distributed as follows: three to each cylinder and one to each crank pin. The cylinder feeds are disposed in such a way as to lubricate the

entire working surfaces of the piston and cylinder wall, while one feed also serves to lubricate the gudgeon pin. The gudgeon pin is hollow, as shown at 8, Fig. 1, and is secured in place by a screwed pin, which is split at the point to prevent the possibility of falling out. At the other end of the hollow gudgeon pin is a plunger with a scoop end, 9, Fig. 1, which is forced against the cylinder wall by means of a spring and collects oil from the lubricating tube already mentioned. The feed for the crank pin is carried through the side of the crank chamber and led to the interior of a revolving ring, 10, Fig. 1, from which the oil is carried by centrifugal force to its destination. The lubrication of the main bearings is effected by means of siphon tubes—an accepted marine practice.

DETAILS OF CONSTRUCTION.

The cylinders are cast separately and are extended below the working barrels to form the upper halves of the crank chambers. The cylinder bodies are completely encircled by ample water jackets, and large access doors, 11, Fig. 1, are provided to facilitate the removal of any deposit which may be formed therein. The cylinder covers and silencers are also jacketed, and the circulating water is supplied by a reciprocating pump, 12, Figs. 2 and 3, placed at one end of the bedplate and actuated by an eccentric, 13, Figs. 1 and 3, on the crankshaft. The pump body is of cast iron lined with brass and is provided with brass non-return suction and delivery valves, spring-loaded escape valve and a large cast-iron air vessel. Access to the suction valve is obtained by removing door, 14, Fig. 3, and to the delivery valve by removing the air vessel. A bilge pump, 15, Fig. 2, of the same size and design, is placed at the other end of the bedplate and similarly operated.

BEDPLATE AND CRANKSHAFT.

The bedplate is cast in one piece, with suitable flanges for bolting down to the seating in the ship, and is provided with machined recesses to receive the main bearing bushes. The main bearing caps, 16, Fig. 1, are readily accessible for removal, and when removed the bushes, 17, Fig. 1, which are made eccentric to the shaft on their outer diameter, can easily be slipped from under the shaft without disturbing any gear whatever. The bushes are of Admiralty quality gun metal with white metal lining.

The crankshaft is a solid forging of Siemens-Martin steel, and is machined all over. Large balance weights, 18, Fig. 1, are secured to the crank webs, and these not only give to the engine an evenness of turning moment and a desirable freedom from vibration, but also serve to reduce the clearance volume of the crank chamber, and thus increase the scavenging efficiency.

The connecting rods are machined-steel forgings fitted with adjustable bearings at top and bottom ends. The top end, 19, Fig. 1, of each rod is of special design to allow of easy adjustment, and the bearing surface is of phosphor-bronze. Examination of the gudgeon pin and bush, also piston and piston rings, is obtained by removing the cylinder cover and drawing the piston upwards after the lower end of the connecting rod has been disconnected. The marine-type crank-pin bush, 20, Fig. 1, is of Admiralty-quality gun metal, with white metal lining, and is accessible for examination or adjustment through the doors, 6, Fig. 3, arranged at the front and the back of the crank chamber.

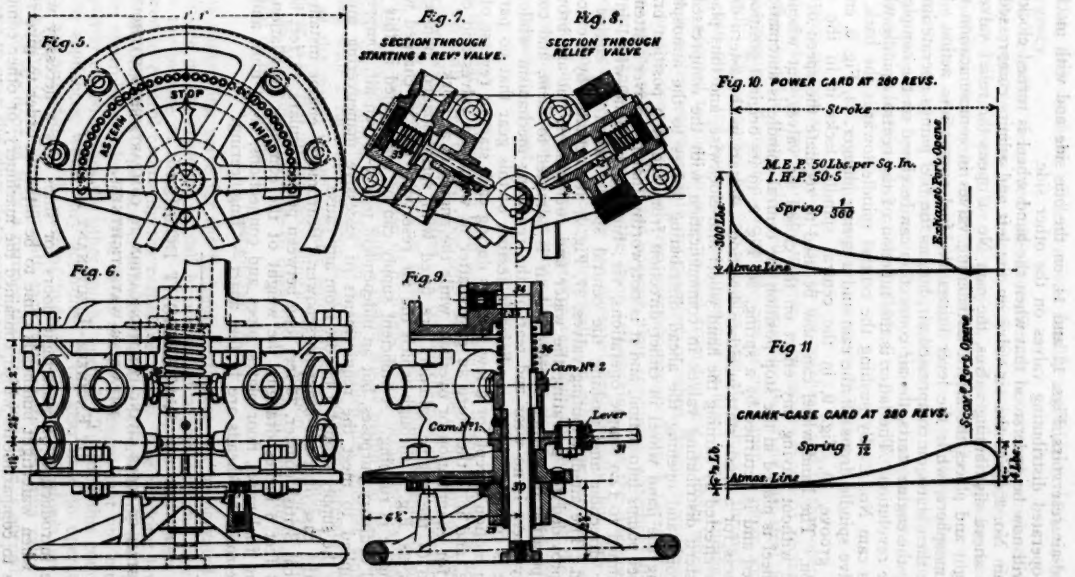
FUEL PUMPS

The four fuel pumps, 21, Figs. 1 to 3, are grouped in pairs at opposite ends of the engine and are operated by the eccentrics, 13, Figs. 1 and 3, previously mentioned in connection with the water pumps, a horizontal

rocking lever, 22, Fig. 3, fulcrumed at its center, being employed to carry the fuel-pump strikers, 23, Fig. 3. The stroke of the strikers is, of course, very much in excess of that to be communicated to the fuel-pump plungers, and consequently the discharge of fuel to the combustion chambers is restricted to a very short period. The idle portion of the stroke is utilized for governing purposes, and as the action of one is identical with the others it will be sufficient to describe one. The pump striker is pivoted on the rocking lever and is free to rotate but for the action of a light spring, 24, Fig. 3, drawing it towards its guide. On the face of the guide a step is formed which throws the striker off its original course, but as this effect is counteracted by the spring already mentioned the engine speed determines whether the striker shall hit or miss the pump plunger. The tension of the spring can be adjusted to cut out at any engine speed, and when that adjustment has been made the following conditions are fulfilled: At all engine speeds up to and including a predetermined limit the fuel pump delivers a definite quantity of fuel during a definite fraction of each engine stroke, the quantity delivered being entirely under the control of the attendant on the starting platform, and when the engine speed exceeds the predetermined limit the pump automatically ceases to operate until the normal speed is regained. When it is remembered that all four fuel pumps are alike, it will be readily understood that the system of governing is very sensitive and quick in action, and it is worthy of note that the tension of the governor springs can be adjusted either individually or collectively while the engine is running. The maximum quantity of fuel to be delivered at full power is determined by a screwed collar, 25, Fig. 3, on the lower end of the pump plunger. By turning this collar to right or left the plunger is raised or lowered relatively to the striker, and consequently the stroke of the plunger is limited, but this adjustment, of course, is only required when tuning the engine to suit a given quality of fuel and should not be used for any other purpose. The ordinary control of the engine speed is effected through a hand wheel, 26, Figs. 1 to 3, placed at the center of the engine. The fuel-pump strikers are mounted on eccentrics in the rocking lever, and these eccentrics are connected by a system of levers and links to a horizontal shaft, 27, Figs. 1 to 3, running along the front of the engine. Any rotary movement of this shaft is therefore transmitted to the eccentrics, and the strikers are thus raised or lowered collectively according to the direction of movement imparted to the horizontal shaft by the hand wheel. Very fine graduations of the engine speed are obtainable in this way. A priming handle, 28, Fig. 1, is fitted to each fuel pump for charging the pipes before starting, and any pump can be put out of action instantaneously by raising the handle to a horizontal position. Should it become necessary for a pump to be put out of action permanently the screwed collar already referred to in connection with tuning meets this requirement, and when thus cut out the suction and delivery valves and springs can be removed for examination by simply disconnecting the delivery pipe and screwing out the plug in the top of the pump barrel.

STARTING AND REVERSING.

Starting and reversing are effected by means of compressed air and the details of the gear, Figs. 5 to 9, show how all the operations of the engine are controlled by a single hand wheel. The hand wheel is mounted on a sleeve, 29, Fig. 9, which also carries a lever and cam No. 1 (Fig. 9), while a central spindle, 30, passing through the sleeve, carries cam No. 2 (Fig. 9). The lever is connected by a link, 31, Figs. 1, 3, and 9, to the fuel pump controlling shaft 27, Figs. 1 to 3, the cam No. 1 opens one of the two relief valves, 32, Fig. 8, and the cam No. 2 opens one of the two high-pressure air valves, 33, Fig. 7. The air valves are directly connected with the com-



DETAILS OF MAIN VALVES AND INDICATOR DIAGRAMS.

pressed-air reservoirs, Figs. 12 and 14, on the one side and with mechanically-operated distributing valves on the other side.

It will now be understood that when the hand wheel is turned clockwise the cam No. 2 opens the air valve on the left and admits compressed air to the ahead distributing valves, the cam No. 1 opens the relief valve on the right and places the astern distributing valves in communication with the atmosphere, while the lever brings the fuel pumps into action. The engine then starts on compressed air, but, as the fuel pumps operate immediately the engine starts, the air connection can be closed on the completion of one revolution. The latter is the function of the central spindle which carries cam No. 2. By pushing the central spindle towards the back, the air-valve spindle drops off the cam and a spring plunger, 34, Fig. 9, entering a groove, 35, Fig. 9, in the central spindle, locks it in the new position. The hand wheel can now be used to regulate the speed of the engine without having any effect on the closed air valves, but whenever the wheel is placed in the stop position the central spindle is automatically released and is returned by a spring, 36, Fig. 9, to its original position in readiness for a fresh start in either direction. A start in the astern direction is effected by turning the hand wheel anti-clockwise, and thus placing the astern distributing valves in communication with the compressed-air reservoirs and opening the ahead distributing valves to the atmosphere. Moving the hand wheel in either direction from the stop position brings the fuel pumps into action, and it is noteworthy that the driver's attention is confined to two simple operations, viz: turning the hand wheel in the proper direction and pushing the central spindle when the engine has started. The air-distributing valves, 37, Fig. 3, are actuated by the eccentrics provided for operating the water and fuel pumps, but are brought into operation only when compressed air is admitted to them, and consequently there is no wear and tear of the reversing mechanism while the engine is running. It is an admirable feature of this gear that no part of it is in motion except for a few seconds during each start or reverse of the engine, and it is therefore an easy matter to withdraw valves and other parts for examination or overhaul while the engine is at work.

Two of the engine cylinders are fitted with spring-loaded, non-return valves, 38, Fig. 1, through which the air reservoirs can be charged while the engine is running. A sufficient supply can be maintained in this way for all ordinary purposes, but an independent power-driven air compressor is supplied to meet the requirements of excessive maneuvering and to renew the supply should it be lost from any cause.

In the case of a 160-brake-horsepower "Beardmore" engine installed in a typical British coaster, 75 feet long between perpendiculars, 18 feet beam, and 8 feet 6 inches draught, the weight of the complete installation, including fuel tanks, floor plates, pipes and connections, is 14 tons, and the engine-room bulkheads are 15 feet apart.—"Engineering."

WATERTIGHT DOORS.*

SYSTEM OF DOORS GIVING COMMUNICATION TO COMPARTMENTS WITHOUT IMPAIRING WATERTIGHTNESS.

By Engineer-Lieutenant I. Toro.†

The introduction of watertight doors for ships was a necessity which arose when watertight bulkheads came to be fitted, in order to give ready access to compartments which contained the machinery, or other compartments to which it was deemed desirable to get access periodically for in-

* Paper read before the Institute of Marine Engineers, London, September, 1917.
† Chilian Navy.

spection, and yet have efficient means readily available to isolate these in emergency.

The necessity demanded the invention of a system which could be readily available when the moment arrived to close the doors. Several plans have been invented for the purpose, and yet there appears room for more—hence that which is now submitted for discussion.

The object of this system of watertight doors is to allow of easy communication from one compartment to another, also to provide means of escape from one compartment to an adjoining compartment without allowing the watertightness of the bulkhead to be impaired.

CONSTRUCTION OF AIRTIGHT CHAMBER.

In this system an opening is cut in the bulkhead, and to this opening is fitted, in a water- and airtight manner, a chamber or cell of rectangular or other suitable shape; this cell is formed by two rigid cast-steel frames of the required dimensions, to enable one or more men to move about in the cell. This chamber has two openings, one at each side, through which men can pass, and each of these openings is fitted with a sliding door (*EE*) adapted to slide in ways to and from the seat, so that, when open, there is a clear entrance into the chamber, or when closed, a wedge-shaped edge on the door enters between correspondingly shaped faces or into a recess in the seat or sill of the opening, thus forming a watertight joint to prevent leakage and render the door self-packing. In the diagrammatic sketch reproduced these doors are operated hydraulically, but they could also be worked pneumatically, electrically or otherwise, in such a way that under no circumstances whatever can both be opened at the same time, one door remaining open when the other is closed, and *vice versa*, or both remaining closed.

OPERATING GEAR.

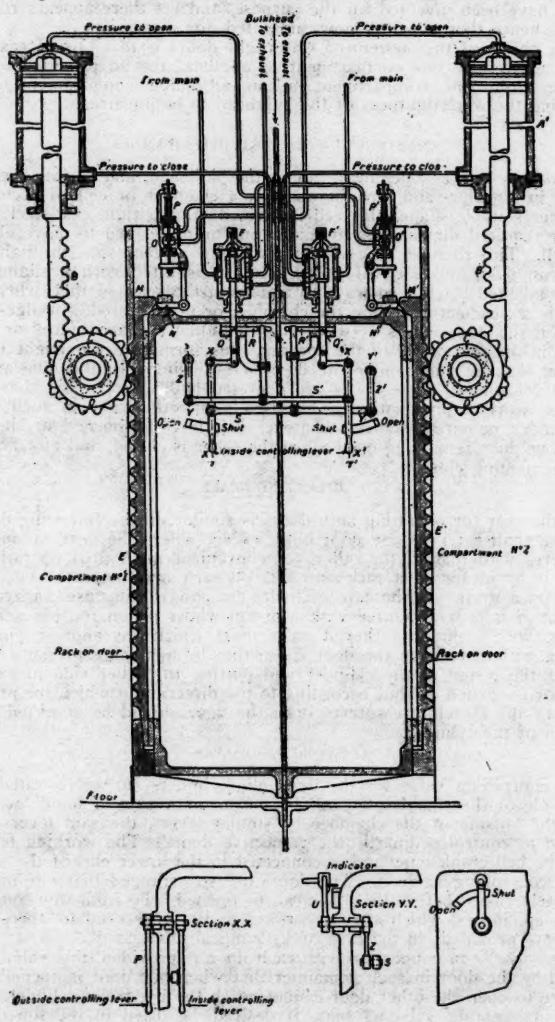
As the gear for operating both doors is similar, in the following description we shall refer to one door only, except where the gear of one door interferes with that of the other. A convenient method of operating the doors is by means of a rack or racks on each door operated by pinions keyed on a shaft. In the case of hydraulic power being used, a hydraulic cylinder *A* is provided fitted with a piston whose piston rod has a rack *B* gearing with a pinion *C* keyed on a shaft which has another pinion *D* gearing with a rack on the door *E*, so that by admitting pressure to one side of the piston in the cylinder and putting the other side to exhaust, the door is opened or shut according to the direction in which the pressure acts. In the sketch, pressure to open the door should be admitted at the top end of the cylinder.

CONTROLLING VALVES.

The controlling valve for the hydraulic cylinders can be operated from the inside of the chamber or cell by means of a pair of hand levers, or from the outside of the chamber by similar levers, the said levers being adapted to control ordinarily the respective doors. The working lever *T* is of the bell crank type, and is connected to the lower end of the spindle of the controlling valve *F*. The doors are so arranged that one must be completely closed before the other can be opened. To fulfil this condition, certain appliances, which vary according to the gear used to operate the doors, are provided; in the case of hydraulically operated doors, an intercepting valve *O* in connection with each door is provided, this valve being worked by the door in such a manner that when one door is opened, fluid pressure to open the other door cannot reach the hydraulic cylinder.

The intercepting valve *O* may, if desired, be fitted in the top of the

chamber, its spindle being connected to a bell-crank lever *N*, one of whose arms is operated by a cam *M* fitted on the top of the door in the rear, and so arranged that only when the door is perfectly closed the cam *M* would move the spindle of the valve *O* to allow a free passage between the con-



DIAGRAMMATIC SKETCH OF DOOR AND CONTROLLING MECHANISM.

trolling valve and the opening end of the cylinder actuating the other door. The chambers of the intercepting valve *O* are connected respectively by pipes with the hydraulic cylinder that operates the other door, and with a port in the chest of the controlling valve that operates the said hydraulic cylinder, so that the pressure fluid, on being admitted by the controlling valves, passes through the pipes by way of the intercepting valve of one door to the hydraulic cylinder of the other door. The chambers of the intercepting valve of the opposite door are similarly connected, so that the opening of one door keeps the other closed, and vice versa. If, therefore, one of the doors be opened, the act of opening will release its intercepting valve spindle, so that such intercepting valve closes by the pressure of the spring *P* and cuts off any admission of pressure fluid into the hydraulic cylinder of the other door. It will be seen, therefore, that if one door be open and the other closed, the operation of the lever to open the closed door would have no effect, because the intercepting valve would be closed and pressure fluid could not reach the hydraulic cylinder to open that door. When both doors are closed, then either lever can be operated to open one door or the other.

LOCKING DEVICES.

To prevent both doors being opened simultaneously there are certain locking devices provided which consist of locking bolts *RR*, coupled either directly or through the medium of rocking arms and levers *SS* with the opposite hand levers, such locking bolts being adapted to lock the valve spindles under certain conditions—that is to say, when both doors are closed, the locking bolts are in the "off" position, but when one or the other of the controlling levers is operated, the locking bolt connected therewith is shot forward into slot *Q* or *Q* cut in the valve spindle of the other controlling valve so as to lock it and prevent it being operated. It will be seen that the locking bolts provide also a device which prevents in any case one door being opened while the other is open, as the slots *QQ* cut in the spindles are only exactly in front of the said bolts *RR* when the levers of the controlling valves are in the shut position—that is to say, when the doors are closed. When one door is open the slot of the controlling valve spindle is moved from its original position, the passage being intercepted by the solid part of the spindle, so that the lever of the other door cannot be moved from the closed or shut position, as the locking bolt connected to it has its travel barred by the spindle of the other controlling valve.

Should one or both of the intercepting valves *OO* not work as intended, this mechanical device provides a further means of preventing both doors being opened at the same time. Moreover, if desired, the use of the intercepting valves could be dispensed with and only the locking bolts used. This one door must always remain shut, and in no case could there be a through passage from one compartment on one side of the bulkhead, or partition, to the other side thereof.

OPERATION OF THE SYSTEM.

The working of this system of doors in order to get from one compartment to another is as follows: The door at one side of the chamber or cell is opened first by means of the outside controlling lever to allow the man or men to gain admittance to the cell. This door is then closed by one of the controlling levers inside the cell. When this is properly closed, but not before, the opposite door can be opened by the other controlling lever, enabling the men to obtain admission into the other compartment. In no case can there be a through passage from one compartment to the other, as one door must always remain shut.

In ordinary circumstances each door may be controlled from its respective side, but as it may happen that the panic-stricken men rushing from a compartment where there is danger to the adjoining compartment might easily forget to shut the door of the safe compartment, thus preventing other people from escaping, an emergency arrangement is applied which permits of the controlling of the doors from the opposite compartment; with reference to the sketch, the lever *S*, besides being connected to the rocking lever that works the locking bolt *R*, is connected also to an arm *Z* pivoted on a shaft which is fitted in a watertight and airtight manner through the side of the chamber; in the outside end this shaft has keyed to it an arm *U*, fitted with a handle and also with an index which works between the positions *open* and *shut*. In this manner the possibility of leaving men locked in a compartment is therefore prevented, as by means of the arm *U* the controlling valve of the opposite door can be worked, and therefore this door can be closed to allow the opening of the other door. If necessary, a device may be applied to give a signal of the completion of the operation of passing from one compartment to the other.

The controlling valves could be of the ordinary type, but the interlocking device might, I think, be applied to some of the patented valves that can be operated from the bridge. In my system there is, however, no need of this control from the bridge, as under no circumstances whatever is there a through passage from one compartment to another; in fact, it may be said, as far as the efficiency of a watertight sliding door permits, that the bulkhead is intact.

As mentioned before, the sketch given is only a diagrammatic one, and the positions of the different parts may be altered to suit circumstances; for instance, in actual practice it will be more convenient to place the controlling valves in the sides of the chamber; in such a case the outside controlling lever would be of the bell-crank type, and would be connected to the spindle of the valve; the inside controlling lever would also be of the bell-crank type, and it would be connected to a cylindrical bar containing the slot for the locking bolts; this slot would be cut in such a position that it would be just in front of the locking bolts only when the controlling valve was in the shut position. The chamber or cell is provided with means for ventilation, and also with lighting arrangements. This system of doors could also be applied when using electrical or mechanical power, but naturally the gear would be somewhat different in construction, though similar in principle. The invention may also be applied as an air lock for submarine vessels and for diving purposes, etc.

In connection with the use of doors in the bulkheads of warships, the naval authorities do not allow their application, although they recognize the difficulties under which the engineering force has to work to obtain full efficiency from machinery placed in isolated places, not always run by careful hands, but the naval authorities consider that where there is a *door* in a bulkhead, there is a *hole*. All this may be applied to single doors, as in the moment of danger a plank of wood, a shovel or any other article might be left at the door, which would prevent it from closing, but with my system such a thing is impossible, as there must always be one door closed, and therefore the bulkhead is always watertight. In merchant ships doors are allowed in bulkheads; with this system of double doors which are *fool-proof*, the safety of such ships would be considerably increased, as, for reasons already stated, the bulkheads would always be watertight, notwithstanding that there would be easy means of communication between the different compartments. I think that this system of doors is *fool-proof* and able to fulfil the demands of always maintaining the watertightness of the compartment, and at the same time permitting safety of entrance to or exit from the compartments.

TANK-STEAMER CONSTRUCTION.*

RECENT GROWTH OF TANK-STEAMER FLEET — STRIKING FEATURES OF DESIGN — STANDARDIZATION.

By Robert W. Morrell, M.E.

The growth of the American tank-steamer fleet since the beginning of the European war has been so remarkable, and the tank steamers now represent such a large and important part of our merchant marine, that the construction of these vessels cannot fail to be a subject of interest in maritime circles.

At the end of the year 1914 the tank steamers under American registry numbered about 80 (exclusive of barges), of 278,837 gross tons and approximately 416,000 tons deadweight, which at the time represented only about 10 per cent. of the total steel tonnage of the United States, excluding the Great Lakes and sailing vessels.

At that time shipbuilding in this country was almost at a standstill and the outlook was very gloomy. So complete was the cessation of activity in the shipyards that this period of depression may be said to represent the line of demarcation between the old shipbuilding era and the new.

Early in 1915 shipbuilding took on a new lease of life, and it is interesting to note that it was the construction of tank steamers which started most of the yards on their new road to prosperity. It was the need for oil carriers which rescued the shipbuilding industry and revived it.

Since the beginning of 1915 there have been built in this country 57 tankers (exclusive of barges), of 371,900 gross tons and 519,200 tons deadweight, thus more than doubling the American tank-steamer tonnage. It will also be noted that the average tonnage of the individual vessels shows a great increase over that of the vessels previously built.

In addition to the extensive building program carried on, a marked addition to the American tanker fleet was made under the provisions of the Panama Canal Act of August 18, 1914, by means of which 40 foreign-built tankers, aggregating over 190,000 gross tons, or 285,000 tons deadweight, have been transferred to American registry.

We therefore have the honor of displaying the American flag from the largest tank ship afloat — the German-built steamer *Jupiter*, now the *Standard*, of 10,073 gross tons and 17,000 tons deadweight. The second largest tanker is a British vessel of 15,500 tons deadweight, while the United States ranks third, having constructed a type of vessel of 15,000 deadweight. It might be mentioned that six vessels of this size have already been turned out in this country, and that four more are nearing completion.

Up to the present date we have lost six tankships through war causes. Two of these were German-built, three British-built and one American-built. They represent a gross tonnage of 26,537 and 38,590 tons deadweight.

The total number of American tankers now in service is therefore about 170, having a total gross tonnage of 823,200 and a total deadweight carrying capacity of 1,180,000 tons. Thus it will be seen that during the recent period of activity the number of tankships has been increased 112 per cent., the gross tonnage increased 185 per cent., and the total deadweight tonnage increased 185 per cent. Not only is this the case, but it is also found that the tank-steamer tonnage is now close to 20 per cent. of the total steel American tonnage (exclusive of the Lakes), as against 10 per cent. at the beginning of 1915, thus indicating that the greatest of the recent progress

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in shipbuilding has been in the construction of tankers, and that the growth of the oil industry has demanded for them a prominent part in our shipbuilding program.

TANK STEAMERS BUILDING.

In addition to this impressive fleet of tankers, we now have under construction in the United States a total of 64 tank vessels, representing 496,680 gross tons and 575,750 tons deadweight, marking the greatest shipbuilding activity in the history of the country. This recent construction on so large a scale could not fail to bring out interesting and instructive features deserving of comment.

The most impressive feature of the modern tanker is the size. Previous to 1915 the average deadweight was 5,200 tons, but the average deadweight of the vessels built in the United States since that time is 9,100 tons.

In 1915 the largest American tankers were the steamers *John D. Archbold* and *John D. Rockefeller,* of 11,500 tons deadweight, but we have since built ten larger vessels, the largest of which attain a deadweight of 15,000 tons. The shipowner's faith in the economy of the large tankers therefore appears to be firmly established. In fact, under certain conditions, the large vessels, trading at ports having limited depth of water and therefore obliged to load part cargoes, have been able to compete with smaller vessels whose light draught permits loading full cargoes.

DEVELOPMENT OF SHELTER DECK TYPE.

Along with the increased size came the development of the shelter-deck type of vessel. The term "shelter-decked type," as commonly used in this country in referring to tankers, applies to a vessel having three continuous steel decks, the uppermost of which is the strength deck. According to Lloyd's Register, a shelter-decked vessel is one in which the uppermost or shelter deck is a light, continuous superstructure, with one or more tonnage openings. If without tonnage openings, it is an awning-decked vessel, while if the upper scantlings are heavier it becomes a spar-decked vessel. The commonly accepted use of the term "shelter-decked vessel" in this country, however, is in reference to any vessel having a continuous weather deck, as opposed to the type having raised forecastle, bridge and poop.

The first shelter-decked tankers built in this country were the sister ships, *John D. Archbold* and *John D. Rockefeller,* of 11,500 tons deadweight each. These vessels had the oil tanks carried up to the upper deck, which is the next deck below the shelter deck. The oil hatches were located on the upper deck, and gas trunks were built around the hatches and extending from upper to shelter deck, to keep the gases from permeating the 'tween-deck space.

Experience has proven these trunks a source of danger to the ships, as the heavy gases accumulate at the hatches and great caution is necessary to prevent men from being overcome when entering the tanks. Apparatus for clearing these spaces of gas has since been installed.

The next step in the development of the shelter-decked type was the carrying of the expansion trunks right up to the shelter deck. This was embodied in the *Charles Pratt* and subsequent vessels of that type with great success.

With this arrangement, the expansion trunk bulkheads extend from the second deck (or main deck or tank deck, as it is variously called) up to the shelter deck, thus making the expansion trunk two decks in height. The summer tanks are arranged outboard of the expansion trunk between the second and upper decks. The oil hatches for both main and summer tanks are located on the shelter deck, and small trunks, extending between the upper and shelter decks, are built for the summer tanks. Outboard

of the expansion trunk, between the upper and shelter decks, are open 'tween-deck spaces above the summer tanks.

The principal advantages of this design over that in which the oil is not carried above the upper deck are as follows:

1. An increase of about 10 to 15 per cent. in the cubic capacity of the tanks is obtained, which is of great benefit when carrying light grades of oil.

2. The vessel can be trimmed, when loaded, with much better results, as the expansion trunk extending through two deck heights gives great scope for the desired ullages. If the vessel has a tendency to trim by the head when leaving port on account of full fuel tanks forward, this can be overcome with the present design by leaving large ullages in the forward cargo tanks and smaller ullages in the after tanks. It is also possible, as fuel is used, to maintain any desired trim by transferring the cargo. The deep trunks allow great leeway in this respect.

3. On account of the oil hatches being on the shelter deck, the danger of getting gas in the 'tween-deck space is avoided.

4. The center of gravity of the cargo is higher, making an easier ship.

5. The expansion trunk and center-line bulkheads, extended up to the shelter deck, provide valuable additions to the longitudinal strength of the ship.

To offset the above, it should be noted that additional steel is required to carry the trunks to the shelter deck and that the scantlings must be increased on account of the increased head of oil. Therefore a slight increase in the hull weight is involved, and, if the vessel is to carry only heavy crude or fuel oils, the extension of the oil space above the upper deck is not advisable. This brings out the fact, which is constantly cropping up, that to design the most economical tankers they must be specialized for particular trades and used in no other trades. The point just mentioned illustrates this—if the vessel is intended to carry light oils, the oil space should be made as large as possible by extending it up to the shelter deck; but if intended to carry only heavy oil, large cubic space is not required and the oil space may terminate at the upper deck.

It should be noted in passing that with the oil hatches on the shelter deck it is not advisable to locate the living quarters amidships directly on the oiltight deck. To overcome this, on all vessels of this type the accommodations have been raised 3 feet 6 inches off the shelter deck, leaving an open space under the house. The house plating is extended down to the deck on the front and sides, but is open for access at the after end. The house with this construction requires particularly secure fastening to the shelter deck.

It is impossible to avoid bringing up some of the valve-operating rods into this space under the house, but these are extended out clear by means of bevel gears.

There have recently been built, or are now building, several shelter-deck vessels in which the oil hatches are located on the upper deck, and the 'tween-deck space is entirely open, without gas trunks, but the majority of the shelter-decked ships are now of the type just described, namely, with the trunk to the shelter deck. This type is highly satisfactory for an all-around tanker.

A recent special design which is worthy of note is embodied in the tanker *D. G. Scofield,* in which the entire 'tween-deck space, between upper and shelter decks, is made suitable for carrying case or barrel oil, in addition to the bulk cargo in the tanks. The oil hatches are on the upper deck and the large cargo hatches are provided in the shelter deck each side of the center strake, which is continuous. Three masts are fitted with complete cargo-handling gear consisting of booms and winches. This is another instance, however, of adaptation of a tanker to special trade.

Such package cargo-handling equipment on the majority of tankers would be useless.

In considering the subject of shelter-decked tankers, the question naturally arises as to what determines whether a vessel shall be built with a shelter deck or a forecastle, bridge and poop deck. The answer is somewhat obscure, but the determining factors seem to be mainly matters of size and of personal preference. The dividing line seems to be in the neighborhood of 10,000 or 11,000 tons deadweight, the vessels above that size being almost without exception shelter-decked vessels.

The main point involved is one of draught. Increasing the size of the vessels it is not desirable to increase the draught in proportion. The shelter-decked type lends itself to carrying an increased deadweight on a limited draught without adopting extreme proportions. The shelter-decked type is also preferable for strength purposes in the larger sizes.

With regard to tonnage openings, a few shelter-decked vessels of this nature have been built, and they have the advantage of small tonnage measurement combined with good structural design due to small length to depth ratio; but the greater depth required tends toward a heavier ship.

For purposes of comparison I have made a rough approximation of the general characteristics of four steamers, each of 11,000 tons deadweight capacity, but of different types, as follows:

1. Shelter-decked vessel with expansion trunk carried to shelter deck.
2. Shelter-decked vessel with oil space up to upper deck only; no tonnage openings.
3. Same as (2), but with tonnage openings.
4. Forecastle, bridge and poop type.

In working out the dimensions the length and beam were kept constant and the depth and draught allowed to vary to suit the different conditions.

The following table gives the results of the calculations, which have not been carried out to any degree of refinement, but which are sufficient to indicate what may be expected of the different types:

COMPARISON OF FOUR TYPES OF TANKERS OF 11,000 TONS DEADWEIGHT EACH.

Type.	L.	B.	Draught.	Depth to upper deck.	Depth to shelter deck.	Displacement.	Weight of ship light.	Cargo capacity, cubic feet.	Gross Tonnage.
1.....	450′	59′	26′ 4″	28′ 0″	35′ 6″	16,270	5,270	483,000	7,900
2.....	450′	59′	25′ 6″	27′ 10″	35′ 4″	15,700	4,700	420,000	7,850
3.....	450′	59′	26′ 0″	31′ 0″	38′ 6″	15,800	4,800	458,000	7,750
4.....	450′	59′	26′ 6″	34′ 0″	.....	16,000	5,000	460,000	7,850

From the table it will be noted that type (1), although a heavy ship, has great cubic capacity, and is therefore best adapted to carrying light grades of oil. Type (2) is a lighter ship, for reasons previously explained, but has small cubic space and is therefore suitable only for heavy oil trade. Type (3), on account of the tonnage openings, requires great depth, resulting in increased weight of hull and increased cubic capacity. Type (4), it will be noted, requires greater draught and is rather heavy, but has plenty of room inside.

It must be remembered that all of these vessels will carry a deadweight of 11,000 tons, and therefore the variations and possibilities with the deadweight fixed, and even with length and breadth fixed, are apparent. The results go to show that the vessel, when designed, must be made to conform to the conditions under which it is intended to operate.

In the matter of cubic cargo space, for instance, let us assume that for a vessel of this size the fuel carried will be 1,050 tons, with 250 tons fresh

water, stores, etc. This leaves 9,700 tons for cargo, which, if heavy fuel oil, occupies 358,900 cubic feet; if refined oil, 436,500 cubic feet; and, if naphtha, 485,000 cubic feet. A study of these figures will readily give an indication of the respective capabilities of the vessels.

The foregoing types were assumed to have the usual sheer line, the middle portion of which, if faired on the ¾-inch lines, would show an easy curve. It must be noted, however, that there is a growing tendency at present to adopt a straight sheer line amidships, and, in fact, there are tankers now under construction embodying this idea.

EFFECT OF STRAIGHT SHEER.

In order to investigate the effect of the straight sheer, the sheer line in type (1) was changed and made straight for 55 per cent. of the length amidships, fairing into the original height at the ends. This change made an increase in the molded depth to shelter deck of 2¾ per cent. The principal objection to the straight sheer is the reduction in height of shelter deck for a space beginning at the stem and extending to a point about 35 per cent. of the length abaft the stem, where the old line crosses the new. The greatest reduction in height occurs between 15 and 20 per cent. of the length abaft the stem. The reduction amounts to only a few inches and is not a serious difficulty.

With the new sheer line the allowable draught will not be noticeably changed, and the difference in hull weight will be negligible. The advantage structurally of the straight sheer is obvious, as it will permit practically universal work throughout almost the entire length of the oil space, and the frames, bulkheads, etc., will be alike for every tank except the end ones.

SUBDIVISION OF CARGO TANKS.

A prominent feature of recent construction has been the subdivision of the cargo tanks into groups for the carrying of different grades of oil. If two different grades are separated by a single bulkhead there is danger that a leak in the bulkhead, permitting a mixing of the grades, will spoil the cargo. The first step was to place the pump room amidships, which, aside from being a most convenient location for pumping, divides the tanks into two groups in which different grades of oil can be loaded without danger of mixing.

This was carried further and cofferdams inserted between the tanks. The 15,000-ton deadweight tankers previously mentioned have ten main cargo tanks divided into three groups with a separate pumping system for each. Several smaller tankers have similar divisions.

This is another example of the incorporation of special requirements for special purposes. As an extreme case, there is one vessel now building on the Atlantic coast which has five cofferdams and a pump room dividing the cargo tanks into four separate blocks. One of the cofferdams, however, is located between the after fuel tank and the fireroom. Many of the modern tankers have this arrangement, which is a wise safety precaution originated on account of fires started by oil in the fireroom due to leaky bunker bulkheads.

Where such cofferdams are built they are provided with tunnels for use in the event of the vessel's burning coal. The tunnels are blanked by oiltight bolted doors on each end, but if used for coal the oiltight doors will be removed and vertical sliding bunker doors fitted. The fuel-oil piping passing through the cofferdam is fitted with a valve on each bulkhead, the valve on the bunker side being controlled from the deck. The cofferdam is also provided with a sea valve controlled from the deck, for flooding in case of fire. Thus, if a fire originates in the boiler room,

it can be isolated from the fuel supply, while if the fire starts in the tanks the propelling machinery can be kept intact.

CYLINDRICAL TANKS.

An important feature of the tank steamer situation is the construction of steamers with cylindrical tanks. Six of this type of vessel have been built or are being built by the Chester Shipbuilding Company. Aside from the saving in hull weight in this type, it has the advantage that the cylindrical tanks can be constructed by people other than shipbuilders. Under the present conditions this fact bears considerable weight. The present design shows a series of tanks the diameter of which is as great as the vessel's framing will permit. A recent design which places smaller tanks on opposite sides of the center line appears to have an advantage with regard to preventing listing of the vessel.

If it is desired to carry cargo or ballast outside the cylinders—that is, between the cylinders and the ship's sides—it is important that the head of water or oil outside the cylinders be not greater than that inside, and for this purpose automatic non-return valves are provided.

The cylindrical tanks are particularly well adapted to conversion jobs, and the few recent conversions of general cargo vessels to tankers have been on this basis. It is a remarkable fact, in view of the great demand for tank tonnage, that so few vessels have been converted to bulk-oil carriers, and rather reflects the skeptical attitude of the shipping community toward converted vessels.

ADOPTION OF ROLLED TEE-BAR SECTION.

Returning to the ordinary tanker and coming down to structural features, it appears that the greatest single item which made for better construction has been the adoption of the rolled tee-bar section for oiltight work. Until recently, tee-bar was not generally rolled for ship use and was unobtainable in proper sizes. Under pressure from shipbuilders and shipowners, however, the mills have adopted this section in several sizes and the advantages of its use in the construction of tankers are generally recognized.

Tee-bar is now being used for the bracket connections of longitudinals to bulkheads, and is replacing single and double-angle connections for this purpose. It has the advantage of being made oiltight much more readily than the angle connections, for the double-angle clips have a habit of leaking at their ends, and the single-angle clip gives difficulty along its heel.

The usual size of tee-bar for bracket connections is 6½ inches by 6½ inches by .45 inch, double riveted in both flanges. Where this replaces a single-angle clip 3½ inches by 3½ inches, single riveted, it is apparent that an equal number of rivets can be obtained in a much shorter tee-bar clip. It is inadvisable, however, even with equivalent rivet connections, to reduce the depth of the brackets on account of using shorter tee-bar clips, and the tendency is, therefore, to use longer tee-bar clips than the riveting requires in order to maintain the effective portion of the brackets.

A further use of tee-bars is for bounding connections on oiltight work. On some of the shelter-decked tankers just built, in which the expansion trunk is carried to the shelter deck, the bar is used to connect the upper-deck plating to the expansion trunk bulkheads, with excellent results. If a single angle were used instead, the heel would require to be calked for the entire length.

Tee-bar is also being used to connect the top of the fore deep tank (under the forward hold) to the fore-peak bulkhead and to the forward cofferdam bulkhead. The shell connection, where the heel need not be

calked, is angle bar, and this butted against the tee-bar and straps fitted. The Sun Shipbuilding Company is using tee-bar for the boundaries of the transverse oiltight bulkheads, and, while the writer has approved this as an emergency measure, there is some question whether double bounding angles are not necessary to maintain proper oiltightness.

The use of the tee-bar for bilge-keel attachments has long been established as the only thoroughly satisfactory connection, and recent experience has indicated that these tee-bars should be at least 9/16-inch thick.

USE OF THE MULTIPLE PUNCH.

One other feature is of great importance for oiltight work, namely, the multiple punch. This has not been used extensively up to the present, but its possibilities are so great that it is believed this machine will soon be in common use in building tankers. The first requirement for oiltight work is that the rivet holes shall be fair—a result which, unfortunately, is not always obtained with the common single punch, especially when operated by a gang on piece work. By using the multiple punch the holes are spaced mathematically correct; every space is exactly the same as every other one, therefore the holes cannot fail to match when the work is erected. If, therefore, the multiple punch will insure fairness of holes, its importance for oil work is obvious. In addition to this advantage, the actual cost of punching, reaming, riveting and testing will undoubtedly be lowered, while the actual laying-off of the work for punching is entirely eliminated where this punch is used.

It must be understood that this type of machine imposes severe restrictions on the rivet spacing, and it is only by painstaking care in the original design and in developing the riveting details that the multiple punch can be used. The chief difficulty is that the spacing in frames or longitudinals must be exactly twice that in seams and butts, which may involve some additional rivets not required by the rules. The spacing in way of bulkheads and transverse frames presents a knotty problem, and it may be that some holes will have to be single punched or drilled.

The multiple punch is, of course, only suitable for work on the dead flat, but if properly worked out it can be used on a very large portion of such work and will result in making the parts universal to a far greater extent than at present.

In view of the large number of tankers recently sunk by submarines, many of which have been in ballast, considerable thought has been given to the question of ballasting in order to give the most favorable conditions if torpedoed. The ballast condition of a tanker is a variable quantity, as it may be altered at will to suit weather conditions or for other reasons. Many claim that if all tanks are filled with water to bring the ship to her load draught, and several tanks are then pierced by explosion, the condition of trim and list will be changed but little, whereas if some of the tanks as to cause the fuel-oil bulkhead to leak into the fireroom, the tanks previously empty, and will list as well, due to the additional water on one side only. On the other hand, if water is carried in all the tanks, obviously some of them must be slack tanks and be even more likely to cause listing.

PROTECTION AGAINST TORPEDOES.

After carefully considering the subject, the writer's conclusions are: First, that no slack tanks should be carried, thereby eliminating this danger from listing. Second, that the vessel be given as much reserve buoyancy as possible, which may be the means of saving the ship if torpedoed in the machinery space; therefore as little ballast should be carried as weather conditions will permit, but whatever is carried should be in full tanks so distributed as to equalize the loading to best advantage. Third,

that gate valves be fitted in the center-line bulkhead near the bottom of each tank, operated from the deck, but left open at sea; thus, if empty tanks are pierced, the water will equalize and prevent listing.

Experience indicates that a torpedo explosion shatters the structure in the vicinity and renders it non-watertight or non-oiltight for a considerable radius. If several tanks are full, the water or oil in them, being practically non-compressible, transmits the shock and destroys the watertightness for a much greater radius. This is another reason for having empty tanks. If the shock of explosion is so transmitted by water in the tanks as to cause the fuel-oil bulkhead to leak into the fire-room, the ship is doomed; therefore anything making this a possibility must be avoided..

STANDARDIZATION.

With reference to the subject of standardizing, this is a matter which has been in the air for some time, and the standardization of tankers has been discussed on many occasions.

So far as the standardization of fittings is concerned, there are great possibilities in this direction. It is understood that the Bethlehem Steel Corporation has already taken steps to standardize fittings in its various plants, and it is to be hoped that this will include fittings for tankers, such as oiltight hatches, shell manholes, tank ladders, cargo-pipe fittings, valve-operating gear, heater coils, gas vents, etc. It would be of great advantage if standard fittings of this nature could be adopted by all the shipyards in the country.

As to standardizing tankers themselves, it would be interesting to note that the 64 tank vessels now under construction in American yards represent nineteen distinct types or designs and are being built in sixteen different shipyards for sixteen different owners, considering vessels building for "foreign account" as being under one owner. At least there were sixteen different owners prior to August 3, at which time all these vessels were requisitioned by the United States Shipping Board, Emergency Fleet Corporation. The construction of these vessels is now in the hands of the Corporation, represented by the district officers; at the urgent request of tankship owners, the district officers have retained the services of the former owners' inspectors, in order that inspection by men thoroughly trained in oiltight work might be assured.

The nineteen types above mentioned range in size from 5,000 tons to 15,000 tons deadweight, and represent almost every possible variation in design. They were designed by competent men to meet specific requirements. Practically every oil company and every shipyard has its own standard type of tanker, or in many cases three or four standard types, all varying in accordance with the different needs of the business and with the different ideas as to how these needs are best met.

It has already been pointed out that special vessels are required for special trades in order to obtain the best results. A vessel designed to carry heavy oil is not suitable for transporting refined oil ,and vice versa. A vessel designed for straight cargo is not suitable for a mixed cargo, but a vessel designed for mixed cargo is needlessly complicated and expensive for shipping straight cargo. Owners trading on the west coast only naturally desire to take advantage of the deep water to adopt wholesome proportions of length to depth, whereas other owners, trading in ports where draught is restricted, must adopt different proportions. Some trades require vessels with fuel capacity for ten days, others for forty days. In many cases large vessels are the most economical, but large vessels cannot be built in all yards and cannot enter all ports.

It is obviously impossible to find a single vessel to meet all requirements, and if an attempt at a compromise is made it will place practically every

oil company at a disadvantage in having to operate vessels which are not quite suitable.

The nearest approach to efficient standardization would be in the adoption of at least four standard designs, consisting of two vessels, a large one and a small one, for the sole purpose of carrying cargoes of heavy oil, and two vessels, a large one and a small one, especially for carrying mixed cargoes of light oil.

In carrying heavy oil, such as fuel oil or crude oil, it is possible to load different kinds in the same vessel without danger of mixing, due to leakage or due to pumping one kind through the same pipe line as another. Therefore there is no need of subdividing the tanks for different grades.

The smaller vessel for this purpose would naturally be of the forecastle, bridge and poop type, and the larger one of the shelter-decked type. In both cases small fuel tanks for bunker use are sufficient, as it is possible for a long voyage to carry half a main cargo tank of fuel, or to carry it in summer tanks. The permanent fuel-oil tanks should consist of a short tank at the forward end of the cargo tanks, and another at the after end, thus enabling the vessel to be trimmed as the fuel is used. With this arrangement no cofferdams whatever are required in the vessel. The pump room in this type of vessel should be located between the after fuel tank and the boiler room, thus separating the oil from the fires, and should contain the fuel-oil pumps and heaters as well as two cargo-oil pumps. A simple system of cargo-oil piping whereby both pumps can draw from the tanks and discharge overboard independently is sufficient. The summer tanks may be fitted with drop valves, connecting them with the main tanks, or they may be piped for fuel oil, as conditions require. All tanks in these vessels should be fitted with heater coils.

For the standard vessels intended to carry mixed cargoes of light-gravity oils, the smaller would be of the forecastle, bridge and poop type and the larger of the shelter-deck type with expansion trunks carried up to the shelter deck. In both vessels rather larger fuel-tank capacity is required. On account of the danger in carrying naphtha next to fuel oil, cofferdams should be provided at each end of the cargo tanks. The cargo pump room should be located amidships, thus dividing the cargo tanks into two groups. For safety, a cofferdam between the after fuel tank and the fireroom should be provided, and the oil-burning apparatus installed in a separate enclosure in the fireroom wing. A fuel pump is required forward to transfer fuel aft from the forward tank. The cargo-oil piping system should permit of the pumping of different grades of oil from the forward and after groups of cargo tanks simultaneously and independently, without the two grades using any piping in common. The cargo piping must not extend into the fuel tanks. The summer tanks should be piped separately. Special precautions against gas must be provided, and the gas vents from the tanks fitted with automatic relief valves. Special ventilation for the pump room and cofferdams must be furnished.

The foregoing outlines briefly the minimum that could be expected in the standardization of tankers. Assuming, however, that such standard designs were prepared and approved, it is not clear what would be gained by throwing out all the present designs and adopting new standard designs. The vessels now building were designed especially for certain definite purposes, best known to their original owners, and it is not likely that standard designs would serve such purposes any better. The ships now building are duplicates of other ships previously built in the same yards, and the experience gained on the previous vessels is an asset not lightly to be ignored. The advantage of having all plans, material orders, templates and patterns for existing vessels places a handicap on standard ships which will take time to overcome, and the result would be to slow up construction instead of speeding up.

Unquestionably, the quickest and most economical means of building additional tankers is to duplicate existing vessels, for which the designs are already perfected and which conform specifically to the needs of the oil business. In this way only can the cheapest and quickest construction and the greatest economy of operations be attained.

FUTURE PROSPECTS OF TANKSHIP CONSTRUCTION.

The future tankship construction is, however, an uncertainty. All of the vessels now under construction are for 1917 or 1918 delivery. When these vessels are completed, the building of tankships will, according to present indications, be at a standstill in this country. So far as we are aware, the United States Shipping Board has not, up to the present time, made any provision for tankers in its shipbuilding program. The Board is promoting the construction of a huge fleet of cargo carriers to transport supplies to our allies and to handle the nation's increasing commerce; but it has entirely ignored one of the most important of the allies' necessities and one of our own principal industries in excluding tankers from its program.

The nation's oil business has advanced with enormous strides and is still advancing. The demand for tank-steamer tonnage is greater than ever, and new vessels will be needed constantly to meet the ever-growing transportation requirements. We have lost many tank ships through war causes and will undoubtedly continue to lose them. In spite of all this, there is no provision for future requirements. Oil companies and shipbuilders have endeavored to obtain the Board's permission to proceed with tankship construction, but up to date this permission has not been granted.

The present situation is, therefore, that the Shipping Board is placing no new orders for building tank ships and will not permit anyone else to do so. This policy can have but one result, and, if immediate steps are not taken, we shall unquestionably be faced by grave shortage of tank-steamer tonnage. This condition is most serious and urgent, and demands the earnest consideration of all shipping men having the nation's welfare, and the cause in the great struggle which it represents, at heart.

RECENT DEVELOPMENTS IN AIR PUMP DESIGN.*

By Mr. E. Jones.

In this paper it is proposed to discuss the merits of some of the more recent developments of one of the auxiliaries used in most power stations. An air pump is a very essential part of the condensing plant, without which any of our modern steam-operated power stations, or, indeed, any machines used for the generation of power using steam for the working fluid, would be practically impossible except for very small units.

It may be advisable at the outset to run very briefly over the various types of condensing plants at present in use, in order to arrive at the cause of inventors turning their attention to the design of new types of air pumps.

There are five types of condenser, viz: (1) evaporative, (2) ejector, (3) barometric, (4) jet, (5) surface.

Evaporative condensers are expensive because of the large cooling surface required to condense the steam, and are only adopted when it is almost impossible to obtain water which can be utilized for cooling purposes. Steam flows in the inside of the tubes, which are grouped together in sections, and all joined up to the header pipes or steam mains. Cooling is obtained by air and the radiation effects caused by allowing a stream of water to continually pass over the tubes. There are a great many joints

* Paper read at the Institution of Engineers and Shipbuilders in Scotland.

in the apparatus, which tend to make this type of condenser unsuitable for modern requirements.

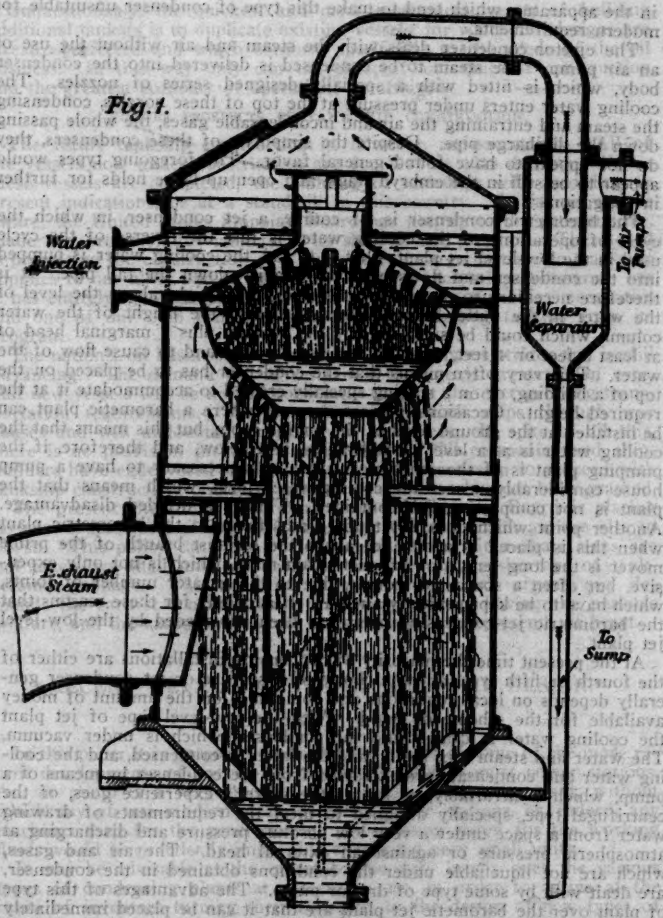
The ejector condenser deals with the steam and air without the use of an air pump. The steam to be condensed is delivered into the condenser body, which is fitted with a specially designed series of nozzles. The cooling water enters under pressure at the top of these nozzles, condensing the steam and entraining the air and incondensable gases, the whole passing down the discharge pipe. Despite the simplicity of these condensers, they do not appear to have found general favor. The foregoing types would appear to be still in the embryo stage, and open up large fields for further investigation.

The barometric condenser is, of course, a jet condenser, in which the cycle of operations of the cooling water is just the reverse of the cycle used in the low-level jet plant—that is to say, the cooling water is pumped into the condenser and flows away by gravity down the tail pipe. It is therefore necessary to place this condenser at a height above the level of the water in the outlet stump equal at least to the height of the water column which could be supported by the vacuum plus a marginal head of at least 3 feet or 4 feet, to allow for pipe friction and to cause flow of the water. This very often means that the condenser has to be placed on the top of a building, or on a staging specially erected to accommodate it at the required height. Occasionally cases crop up where a barometic plant can be installed at the ground level of the engine room, but this means that the cooling water is at a level of at least 34 feet below, and therefore, if the pumping plant is of the centrifugal type, it is necessary to have a pump house considerably nearer the cooling-water level, which means that the plant is not compact or self-contained. This is a decided disadvantage. Another point which does not tend to add favor to the barometric plant when this is placed at a distance above the exhaust branch of the prime mover is the long length of exhaust-steam main, which is not only expensive, but often a source of trouble, due to the greater number of joints, which have to be kept perfectly airtight. It is chiefly for these reasons that the barometric jet-condensing plant has been superseded by the low-level jet plant.

At the present time the majority of condensing installations are either of the fourth or fifth type, and the choice of a surface or jet condenser generally depends on local conditions and sometimes on the amount of money available for the scheme on hand. With the low-level type of jet plant the cooling water is drawn into the condenser, which is under vacuum. The water and steam mix freely, and the latter is condensed, and the cooling water and condensate are extracted from the condenser by means of a pump, which is invariably, as far as the writer's experience goes, of the centrifugal type, specially designed to meet the requirements of drawing water from a space under a very low absolute pressure and discharging at atmospheric pressure or against an external head. The air and gases, which are not liquefiable under the conditions obtained in the condenser, are dealt with by some type of dry-air pump. The advantages of this type of plant over the barometic jet plant are that it can be placed immediately below the turbine, thereby reducing the possibility of air leaks. The whole equipment is self-contained and compact, and can be operated and regulated at very short notice by the engine-room attendant. Figs. 1 to 4 show a typical barometric jet plant and low-level multiple-jet plant of the "Leblanc" type.

Perhaps the most important type of condensing plant is the surface condenser, inasmuch as it is used almost invariably for marine installations, and in a lesser degree for large power units, the reason for this being that the condensate is an important item, as it supplies a source of excellent boiler-feed water, and also under the same external conditions the power

Fig. 1.



required to operate surface-condensing plant is generally less than for a jet plant. The design of modern surface condensers, as far as the condenser itself is concerned, has not changed materially for the last thirty years or so, so that there is no need to go further into details on this point. The cooling water is usually supplied by means of a centrifugal pump, except in cases where a siphonic action can be utilized, or where existing pumping plant is capable of taking care of the requirements of the condenser. A condensate pump is necessary to deliver the condensate from the condenser to the boiler-feed tank, and a dry-air pump to deal with the air and unliquefiable gases which find their way to the condenser. Alterna-

tively, an Edwards air pump can be adopted to deal with both the condensate and air. If the disposition of the various parts of the installation demand it, a hot-well will also be required.

Table I shows the relative costs and requirements of the last three types of condensers which have been considered. The conditions which have been assumed are the same for each case, and are as follows: Steam quantity, 40,000 pounds per hour, vacuum 28½ inches (barometer 30 inches) =95 per cent. of the barometer, cooling water 60 degrees F., prime mover,

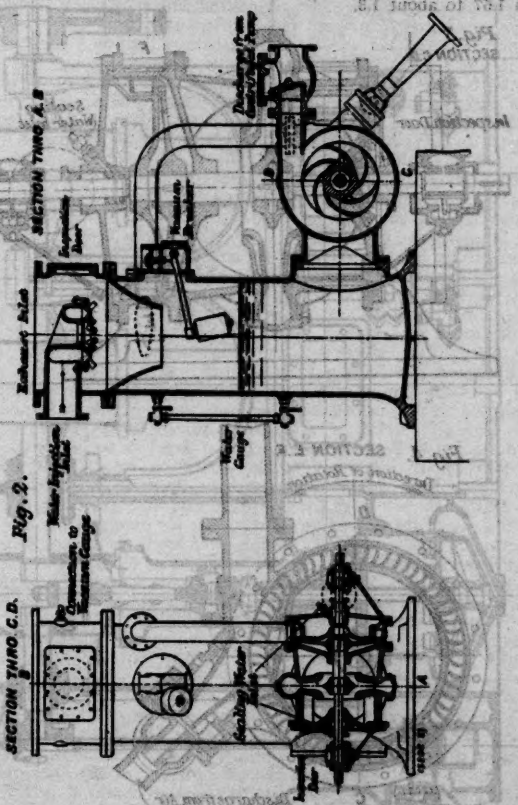
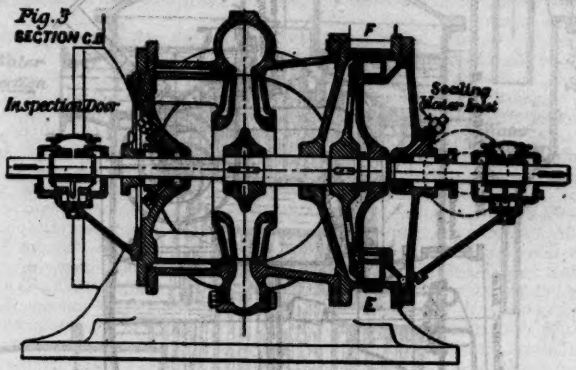
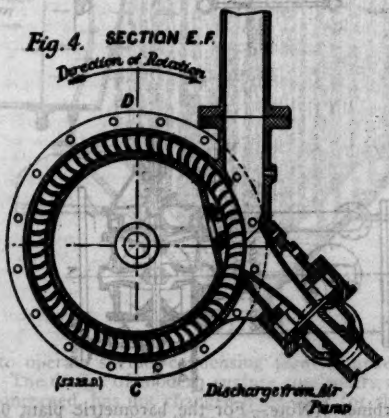


Fig. 2.

high-pressure steam turbine. (Note.—For the barometric plant 0.25 inch has been added to the vacuum to allow for the drop between the turbine exhaust flange and the condenser.) It should be noticed here that under the present abnormal conditions surface condensers are in a particularly unfortunate position, owing to the cost of the materials used for the tubes and tube plates, which is approximately 27½ per cent. of the value of the whole equipment. The price of tubes today is 19d. per pound, whereas in

1914, just prior to the commencement of the war, the price was 7⅞*d.* per pound, so that the increase in the cost of tubes is 240 per cent., and the increase in the cost of plates is approximately the same. Therefore, if times were normal, surface-condensing equipment would appear in a much better light, and although it is not proposed to suggest an accurate figure for surface plant with tubes at 7⅞*d.*, it is clear that a reduction of from 500*l.* to 600*l.* should be possible in the case given above. This would bring the cost ratio of the surface plant, when compared with multi-jet plant, from 1.57 to about 1.3.





With the earliest types of surface-condensing equipment it was usual to use one pump for removing both the condensate and air from the condenser. This pump was known as a "wet" air pump, and a good example of this type is the Edwards air pump, one of the most efficient of its class, shown in section in Fig. 5. The chief advantages of this type of pump are: (1) low power required for driving; (2) positive action and consequent

stability; (3) ability to cope with excessive air leakages. While the Edwards pump is still an excellent pump for units up to, say, from 3,000 kw. to 4,000 kw., it must be remembered that, with the ever-growing size of power units, its disadvantages should be kept in view. For large units with Edwards pumps it is necessary that they should run at a very low

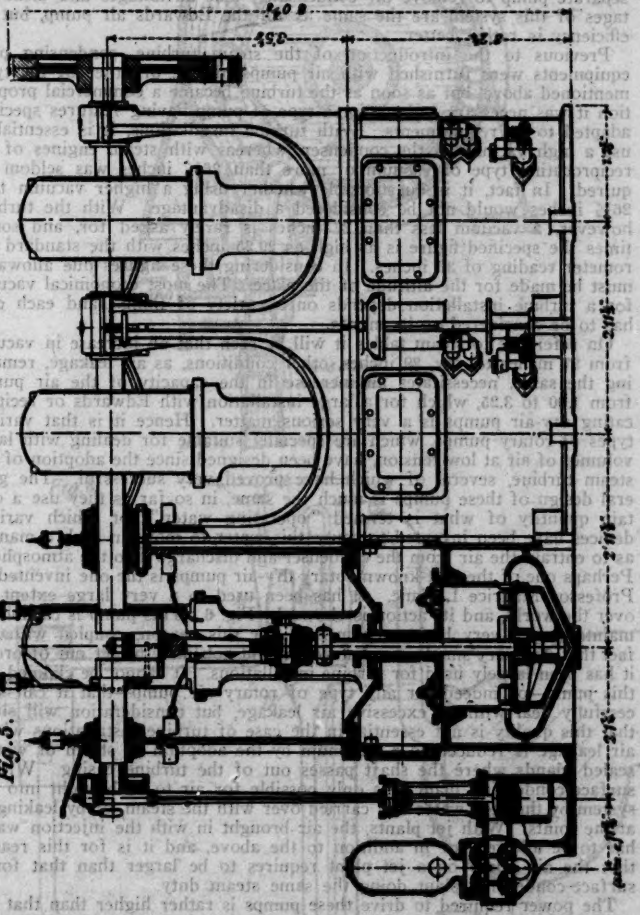


Fig. 5.

speed, and consequently they are very cumbersome and take up a large amount of floor space.

With jet plants the Edwards pump is sometimes used as a dry-air pump. It is necessary, of course, to provide a small quantity of water for sealing

purposes. Volumetric efficiency in this pump varies considerably with the degree of vacuum required, and decreases as the vacuum increases from about 50 per cent. at 3½ inches absolute pressure to 18 per cent. at 1 inch absolute pressure. Another system is that in which a dry-air pump of the reciprocating type is used to remove the air and uncondensed gases and a separate pump to remove the condensate. The advantages and disadvantages of this system are the same as for the Edwards air pump, but the efficiency is rather better.

Previous to the introduction of the steam turbine, condensing plant equipments were furnished with air pumps of one or other of the types mentioned above, but as soon as the turbine became a commercial proposition it was necessary to look for a type of pump having features specially adapted to its requirements. With turbine installations, it is essential to use a high vacuum in the condenser, whereas with steam engines of the reciprocating type of vacuum of more than 26½ inches was seldom required. In fact, it is questionable whether using a higher vacuum than 26½ inches would not be considered a disadvantage. With the turbine, however, a vacuum less than 27 inches is rarely asked for, and sometimes the specified figure is as high as 29.25 inches with the standard barometer reading of 30 inches. In considering these figures due allowance must be made for the altitude of the place. The most economical vacuum for a turbine installation depends on a variety of things, and each case has to be considered on its merits.

On reference to steam tables it will be seen that an increase in vacuum from 27 inches to, say, 29 inches, other conditions, as air leakage, remaining the same, necessitates an increase in the capacity of the air pumps from 1.00 to 3.25, which for a large installation with Edwards or reciprocating dry-air pumps is a very serious matter. Hence it is that various types of rotary pumps, which are specially suitable for dealing with large volumes of air at low tension, have been designed since the adoption of the steam turbine, several of which have proved very successful. The general design of these pumps is much the same, in so far as they use a certain quantity of what is termed "operating water," for which various devices have been invented to cause this water to move in such a manner as to entrain the air from the condenser and discharge it to the atmosphere. Perhaps one of the best-known rotary dry-air pumps is the one invented by Professor Maurice Leblanc. It has been used to a very large extent all over the world, and its action is shown by Fig. 6. This pump is capable of maintaining a very high vacuum, and for this reason, coupled with the fact that it is very simple in construction and not likely to get out of order, it has been largely used for turbine installations. It cannot be claimed for this pump—or indeed for any type of rotary air pump—that it can successfully deal with an excessive air leakage, but consideration will show that this quality is not essential in the case of turbine installations where air leakage is reduced to a minimum by the adoption of steam or water-sealed glands where the shaft passes out of the turbine casing. With a surface-condensing plant it is only possible for air to be brought into the system by the feed water and carried over with the steam, or by leaking in at the joints. With jet plants, the air brought in with the injection water has to be allowed for in addition to the above, and it is for this reason that the air pump on a jet plant requires to be larger than that for a surface-condensing plant doing the same steam duty.

The power required to drive these pumps is rather higher than that required for an Edwards or other good type of reciprocating air pump, and consequently a good deal of attention has been paid recently to another type of pump which would incorporate the simplicity and compactness of the rotary pump and the low-power consumption of the Edwards and other reciprocating pumps. The general trend of thought seems to have been in

Fig.6.

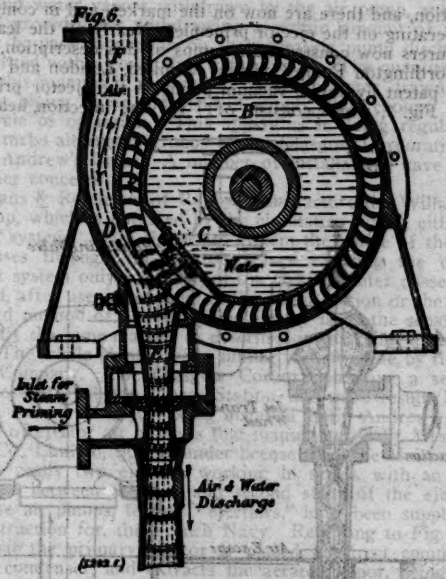
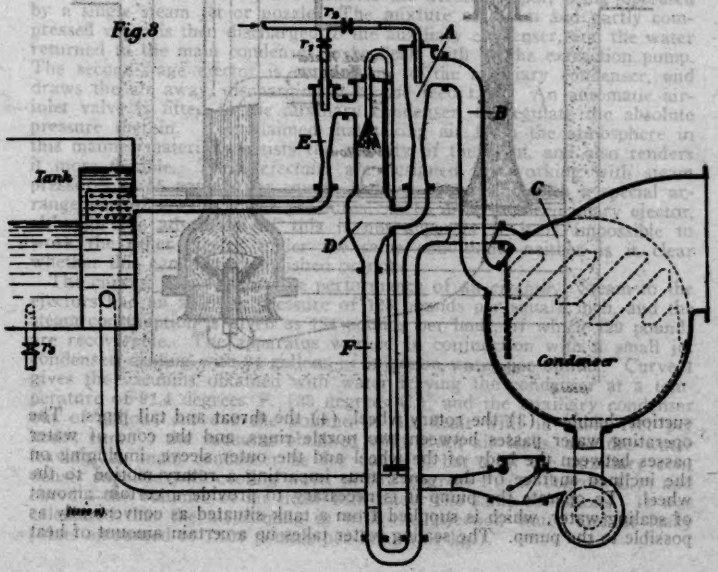

F
Air
B
C
Water
Inlet for Steam Priming
Air & Water Discharge

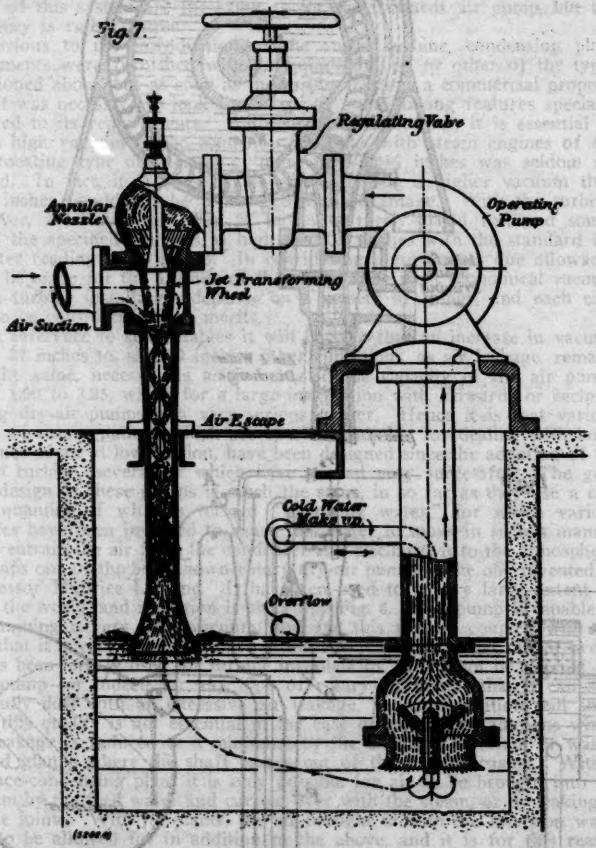
Fig.8


A
B
C
D
E
F
Tank
Condenser

one direction, and there are now on the market, and in commercial use, air pumps operating on the ejector principle. Nearly all the leading condenser manufacturers now construct air pumps of this description.

The Worthington Pump Company, Limited, London and Newark, manufacture a patent hydraulic vacuum pump on the ejector principle, as illustrated by Fig. 7, which consists of: (1) the injection head, (2) the air-

Fig. 7.



suction chamber, (3) the rotary wheel, (4) the throat and tail pipes. The operating water passes between two nozzle rings, and the cone of water passes between the body of the wheel and the outer sleeve, impinging on the inclined surface of the vanes, thus imparting a rotary motion to the wheel. To operate the pump it is necessary to provide a certain amount of sealing water, which is supplied from a tank situated as conveniently as possible to the pump. The sealing water takes up a certain amount of heat

from the air and water vapors withdrawn by the air pump, and a piping arrangement is provided for withdrawing a certain amount of this water by means of a by-pass connection on the operating pump discharge, this by-pass being fitted with a controlling reflux valve. The quantity of water withdrawn in this manner is replaced by make-up water drawn from the circulating inlet piping or an independent supply, thus cooling the water used in the cycle of operation. This apparatus is doing regular service on one of the turbo-alternator groups at the Glasgow Corporation Power Station at St. Andrew's Cross. A number of installations have also been supplied to other concerns.

Messrs. Willans & Robinson, of Rugby, manufacture the Willans-Muller ejector air pump, which is operated by the circulating water, either on the series or shunt system. With the series system the whole of the circulating water passes through the ejector before enternig the condenser. With the shunt system only a portion of the cooling water passes through the ejector, and, after use, is returned to the pump suction or the source of supply. A third method of operating this ejector is by the separate-pump system, in exactly the same manner as described in referring to the Worthington pump. The whole plant is very similar to that made by the Worthington Pump Company. The Glasgow Corporation have a set of this apparatus at work at Pinkston Power Station, and good results have been obtained, and a second set is just being installed at St. Andrew's Cross.

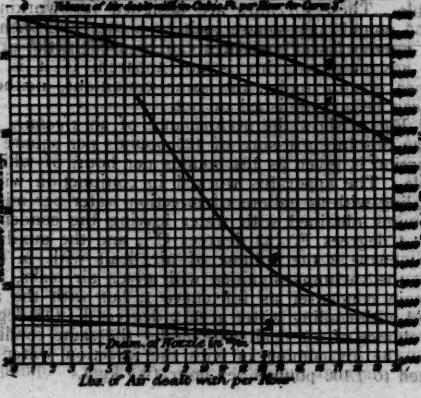
Another type of ejector air pump is that manufactured by Messrs. Hick, Hargreaves & Co., Limited, Bolton, under license from the Maison Breguet, Paris. This is really two ejectors working in series, with an auxiliary condenser placed between the first and second stage of the ejectors. A number of these air pumps, termed "ejectairs," have been supplied to, or are under construction for, the French Navy. Referring to Fig. 8, it will be observed that the primary ejector is placed in direct communication with the main condenser, and extracts the aerated vapor, being operated by a single steam jet or nozzle. The mixture of steam and partly compressed vapor is then discharged to the auxiliary condenser, and the water returned to the main condenser to be dealt with by the extraction pump. The second-stage ejector is coupled up to the auxiliary condenser, and draws the air away, discharging it to the feed tank. An automatic air-inlet valve is fitted to the auxiliary condenser, to regulate the absolute pressure therein. It is claimed that taking air from the atmosphere in this manner materially assists the stability of the plant, and also renders it more flexible. These *ejectairs* are designed for working with steam pressures at 55 pounds per square inch or above, and with a special arrangement of nozzles lower pressures can be used in the primary ejector, although the advantage of this is not apparent if it is impossible to work the other ejector under the same conditions, neither is it clear whether this can be accomplished or not.

The curves, Fig. 9, show the performance of an *ejectair*. Steam to the ejectors had an absolute pressure of 125 pounds per square inch, and the steam consumption is given as 194 pounds per hour, of which 129 pounds are recoverable. The apparatus worked in conjunction with a small jet condenser, dealing with 94 gallons of injection water per minute. Curve 1 gives the vacuums obtained with water leaving the condenser at a temperature of 91.4 degrees F. (33 degrees C.), and the auxiliary condenser out of action; curve 2 the volume of air dealt with in cubic feet per hour; curve 3 the vacuums obtained with given air leaks, and the water leaving the main condenser as for curve 1, but with the auxiliary condenser supplied with cooling water at 66.2 degrees F. (18 degrees C.); and curve 4 the volumes of air dealt with under the same conditions. It was calculated that the air coming in with the injection water and at leaky joints amounted to 1.102 pounds per hour (0.5 kg.).

TABLE I.—*Relative Cost and Requirements of Barometric, Low-level Jet, and Surface Condensers.*

Type of Plant.	Consisting of—	Approximate net Weight of Apparatus.	Horse-Power required to Drive Condenser and Auxiliaries.	Present-Day Costs.	Equivalent Ratio.
		Tons.	b.h.p.	£	
Barometric	Condenser, staging, air pump, injection pump, driving motor, switchgear, air and tail piping, main exhaust piping and sluice valve, auto-exhaust valve. Complete erection	40	89	2,700	1.286
Low-level jet	Condenser, water extraction and air pumps, driving motor, switchgear, main sluice valve, adapting and expansion pieces, auto-exhaust valve. Complete erection	22	87	2,100	1.00
Surface	Condenser, rotary air pump, extraction pump and circulating pump mounted on common bedplate with driving motor and switchgear. Necessary interconnecting pipes, main exhaust-steam sluice valve, expansion piece, adapting piece, automatic atmospheric valve. Complete erection	30	78	3,300	1.571

Fig. 9. BREGUET EJECTAIR
THEORETICAL VACUUM WITH WATER AT 51·6° FAH.



The British Westinghouse Electric and Manufacturing Company, Limited, Manchester, and the firm of Messrs. The Mirrlees Watson Company, Limited, Glasgow, manufacture an ejector air pump, under license from the Société Anonyme Westinghouse, Paris and Le Havre, which is another invention of Monsieur Maurice Leblanc. It is the outcome of many months of arduous research work, during which time innumerable difficulties were surmounted by the inventor, with the result that a really first-class ejector air pump has been evolved. Figs. 10 and 11 show the general arrangement of this apparatus. It will be noticed that the pump is arranged to work in two stages, and the steam is admitted to the second stage of the ejector by opening the stop valve C. Immediately C is opened steam fills the annular space behind the nozzle plate, and finds its way into the throats of the group of nozzles attached to this plate; it then passes along the steam pipe

Fig. 10.

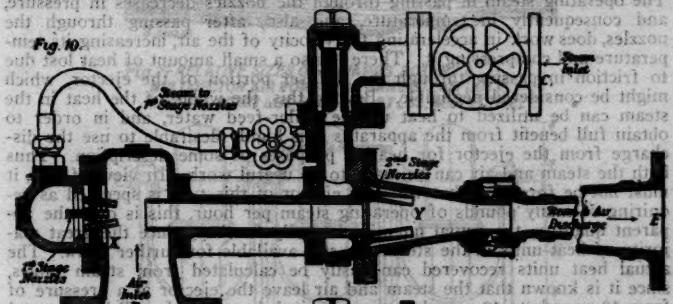
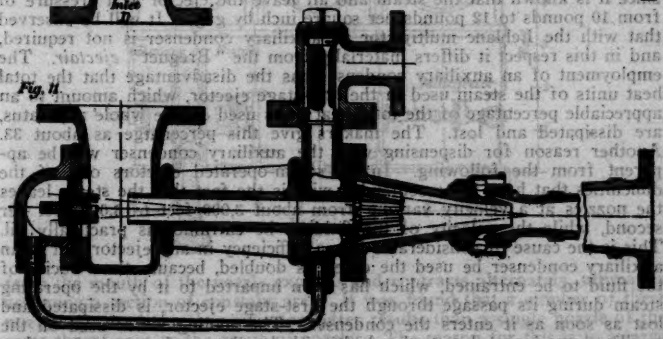


Fig. 11.



which supplies the first-stage nozzles, which are also attached to a nozzle plate. The supply of steam in this set of nozzles is controlled by the stop valve on the steam-supply pipe. The pump is connected to the condenser at the branch D, which is the air-inlet branch. At the entrance to each of the steam spaces fine wire-gauze strainers are fitted to prevent any foreign matter, which may have primed over with the steam from the boilers, from entering the nozzles, thereby intercepting any stoppage in the nozzle throats, and consequently a loss of vacuum. These nozzles are efficiently locked to the nozzle plates. The mixture of air and steam is discharged at the mouth of the divergent cone Y, and led away to the boiler-feed tank, so that the heat units contained in the operating steam can be reclaimed by heating the feed water. To start the pump to work

it is only necessary to open up the steam valve C, and the vacuum will at once commence to increase in the condenser or other vessel to be evacuated. When the vacuum gage becomes stationary the second-stage steam-inlet valve is opened up to bring the vacuum to a maximum. A very important feature in this pump is the absence of moving parts. The simplicity of the apparatus is even more remarkable than that of the Leblanc rotary pump.

The advantages claimed are as follows: (1) extreme simplicity, (2) the small amount of energy required for operating purposes, (3) the high efficiency obtained, (4) ease with which starting can be effected, and the small amount of attention required whilst at work, (5) ability to produce the highest possible vacuums, (6) stability. In scanning these claims we can pass over the first, which, as already mentioned, is obvious; there are simply two steam valves to open. The second deserves some consideration. The operating steam in passing through the nozzles decreases in pressure, and consequently in temperature, and also, after passing through the nozzles, does work in accelerating the velocity of the air, increasing its temperature and compressing it. There is also a small amount of heat lost due to friction in passing through the diffuser portion of the ejector, which might be considered negligible. Beyond this, the whole of the heat in the steam can be utilized to heat up the boiler-feed water, and in order to obtain full benefit from the apparatus it is highly desirable to use the discharge from the ejector for heating purposes of some description. Thus both the steam and air can be made to do useful work. In view of this, it must not be forgotten that when an ejector of this type is specified as requiring so many pounds of operating steam per hour, this is only the apparent quantity, the actual quantity is really far less, since the great majority of heat units in the steam are still available for further work. The actual heat units recovered can easily be calculated from steam tables, since it is known that the steam and air leave the ejector at a pressure of from 10 pounds to 12 pounds per square inch by gage. It will be observed that with the Leblanc multijector an auxiliary condenser is not required, and in this respect it differs materially from the "Breguet" *ejectair*. The employment of an auxiliary condenser has the disadvantage that the total heat units of the steam used in the first-stage ejector, which amount to an appreciable percentage of the total heat units used on the whole apparatus, are dissipated and lost. The makers give this percenatge as about 33. Another reason for dispensing with the auxiliary condenser will be apparent from the following. In all steam-operated ejectors one of the difficulties that have to be contended with is the fact that the steam leaves the nozzles at a velocity varying from about 3,000 feet to 3,600 feet per second, while the velocity of the fluid to be entrained is practically nil. This is the cause of considerable loss of efficiency in any ejector, but if an auxiliary condenser be used the defect is doubled, because the velocity of the fluid to be entrained, which has been imparted to it by the operating steam during its passage through the first-stage ejector, is dissipated and lost as soon as it enters the condenser. The cooling water used on the auxiliary condenser has to be dealt with by the condensate pump, thus increasing the power absorbed by the plant. When working with surface condensers this water must be of good quality, as it has to be returned to the boilers.

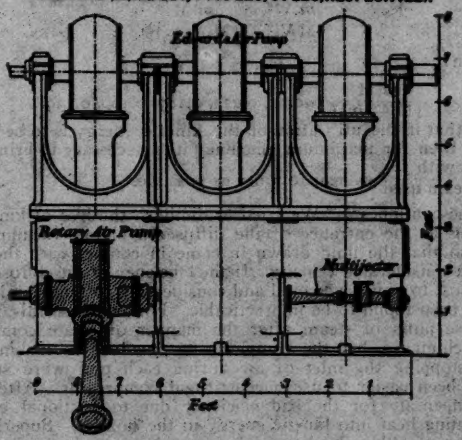
The third claim relates to efficiency. It is well known that ordinary single-stage ejectors only work well when the compression ratio is as 1:7, and it is partly for this reason that the Breguet Company have introduced the auxiliary condenser, so that the vacuum obtained in this condenser is about 25.6 inches with the barometer at 30 inches, the compression being approximately as 11:76, or, roughly, 1:7. The overall efficiency of this plant is, therefore, apparently still further reduced, because air is admitted from the atmosphere into the auxiliary condenser, which is under a

vacuum of 25.6 inches, and this, together with the air from the condenser, has also to be ejected by the secondary ejector to the atmosphere.

When Professor Leblanc set out to design his ejector he foresaw the possibility of using an intermediate condenser, but he also appreciated its disadvantages, and decided to do without it if at all possible. At the same time he knew that it was essential to use two stages in order to get a stable and efficient ejector. With this end in view certain steam nozzles were designed on the lines of the formulæ of Professor Rateau, and the action of the steam issuing from these nozzles when under high vacuum was directly observed. The result is shown by Fig. 13. The steam issuing

Fig. 13.

Fig. 14. RELATIVE SIZES OF EDWARDS, ROTARY & MULTIJECTOR AIR PUMPS FOR SAME STEAM DUTY. WEIGHTS 20832 LBS, 4480 LBS, 97 LBS, RESPECTIVELY.



from the mouth of the nozzle expands and contracts alternately, ultimately assuming a section of constant area. It was found that a number of these nozzles grouped together gave far better results than a single nozzle of the same throat area as the group of nozzles. The reason for this is to a large extent due to the fact that the alternate increasing and decreasing of the cross-sectional area of the steam stream is minimized by the contact of one steam stream with the next, when groups of nozzles are employed, and this helps considerably to increase the surface available for the entrainment of the air and gases. This entrainment is carried on mainly by friction, and it will be seen that if an appreciable amount of gas has to be dealt with, the frictional surface exposed to the gas has to be as large as possible. It is also inversely proportional to the density of the gas or fluid.

The number and size of the nozzles depends entirely on the space available in the diffuser, and keeping within the limits of workshop prac-

tice. The smallest number employed by the Mirrlees Watson Company is three, each of which has a throat diameter of 1 mm. These are first-stage nozzles. On the largest size of pump, and in the second stage, the number of nozzles is 30, and these have a throat diameter of 5.3 mm. when using operating steam at 90 pounds per square inch.

Professor Leblanc, in his paper of 1911 to L'Association Technique Maritime, says that "the operating steam entrains the air by friction. During entrainment it is the velocity of the steam which is utilized, and not its kinetic energy."

Calling M the weight of operating steam used per second, V its velocity at the outlet of the nozzles, m the weight of air drawn in per second, and W the velocity of the mixture of air and steam, then—

$$M V = (M + m) W.$$

The ratio of the kinetic energy, $\frac{m W^2}{2}$, of the air drawn in to the kinetic energy, $\frac{M V^2}{2}$, contained in the operating steam as it comes out of the nozzles can therefore be stated as

$$\frac{m W^2}{M V^2} = \frac{m M}{(M + m)^2}$$

so that when $\frac{m}{M} =$ 1 $\frac{1}{2}$ $\frac{1}{3}$ $\frac{1}{4}$ $\frac{1}{5}$

$$\frac{m M}{(M + m)^2} = 0.25,\ 0.222,\ 0.187,\ 0.160,\ 0.139.$$

This shows that if the utilization of the kinetic energy is to be the basis of the design, then for maximum efficiency it is necessary to bring the ratio $\frac{\text{air dealt with}}{\text{operating steam used}}$ as near to unity as possible.

M. Leblanc continues: "We tried to diminish the loss of kinetic energy by producing at the entrance of the diffuser a higher vacuum than was necessary, so that the fluid drawn in came in contact with the operating steam with a considerable velocity. If the efficiency of the diffuser could be brought almost to unity, we could add considerably to the overall efficiency, but this has been found to be impracticable. Following on this, it was suggested to use puffs of steam after the manner of steam coming out of locomotive chimneys, but the complications involved in making arrangements for stopping the inlet of air during each puff were such that it would have been easier to use a centrifugal compressor. Afterwards we tried to compensate for the bad efficiency due to frictional entrainment by transforming heat into kinetic energy in the nozzles. Superheating the operating steam, although so useful for turbine work, is, however, not good for an ejector, because it is more difficult to effect compression in the diffuser, which outweighs the advantages obtained in the nozzles. The next scheme was to use hot water in the nozzles, but this also proved unsuccessful."

After numerous other trials it was decided that entrainment by friction was most economical, and various types of diffusers and different groupings of nozzles were experimented with, until the present ejector, as shown in Figs. 10, 11 and 12, was decided to be the most suited for condenser work. To go through the various stages in detail which led up to this design would take up too much time. With the form of ejector adopted it has been found that the efficiency of the nozzle is on an average 85 per cent., whilst that of the diffuser is 70 per cent. It will be seen that this ejector agrees very well with the ideal ejector which Professor Leblanc had in his mind. The first stage, which consists of a small group of nozzles, serves a triple purpose, inasmuch as it effects a certain amount of compression, heats up the entrained air, and gives it considerable velocity,

and consequently an increase in momentum. The second stage has a larger number of nozzles, and it is here where the major portion of the work is done, the air being compressed from approximately 26 inches vacuum up to something more than atmospheric pressure. To be more accurate, the steam used in the first stage is about 5 per cent. of the total.

In support of the fourth advantage which this air pump is supposed to possess the following figures were obtained on the French torpedo-

Fig. 15. AIR DEALING CAPACITY OF VARIOUS MIRRLEES LEBLANC MULTIJECTORS.

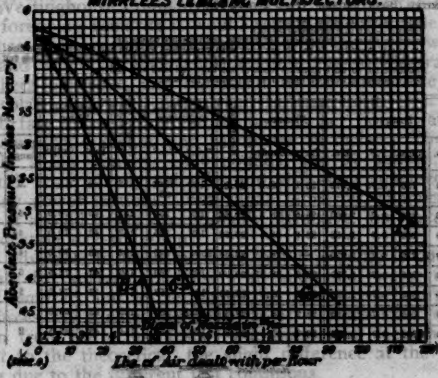
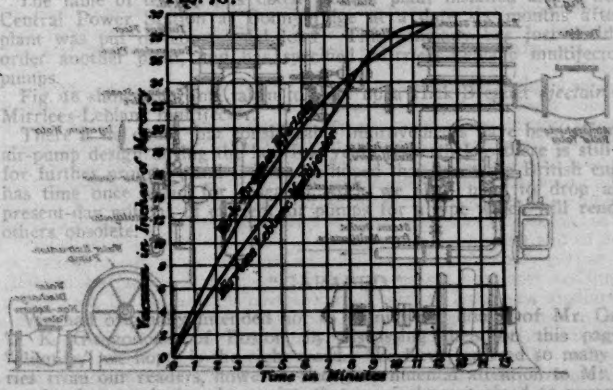


Fig. 16.



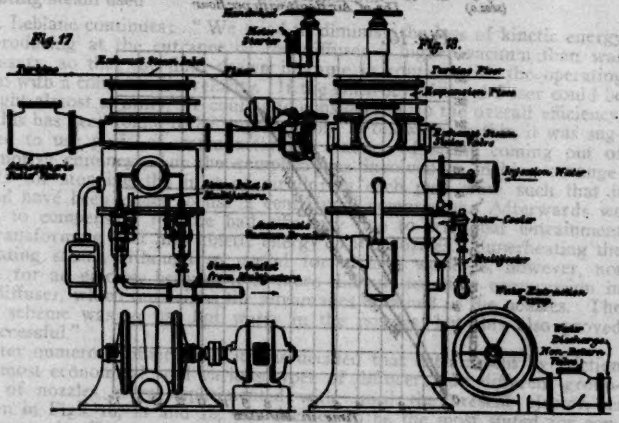
destroyer *Boutefeu*: The turbines were stopped, but steam was on the glands. The volume to be evacuated was about 635 cubic feet. After 1 minute the vacuum was 6-5/16 inches, 3 minutes 15 inches, 3 minutes 22-13/16 inches, 4 minutes 25-9/16 inches, 5 minutes 26¼ inches, and 6 minutes 27-9/16 inches. The theoretical vacuum corresponding to the temperature of the water, 67.1 degrees F., viz: 28¼ inches, was attained in 11 minutes. It was also arranged later to allow certain known air leakages

to enter the condenser. With a 5-mm. nozzle, which passes 36.2 pounds of free air per hour, the vacuum dropped only ⅛ inch. With a 15-mm. nozzle, which is equivalent to 326 pounds of air per hour, the vacuum was 21¼ inches. With an inch cock full open it took 11 minutes for the vacuum to fall to 12¼ inches, at which figure the mercury column remained steady. On closing the regulating valve below the nozzle the vacuum at once rose and attained the maximum almost immediately.

TABLE II.—OFFICIAL TEST OF NO. 18 M. J. CONDENSING PLANT WITH MOTOR-DRIVEN WATER-EXTRACTING PUMP, 1-SIZE "G" AND 1-SIZE "I" MULTIJET AIR PUMP INSTALLED AT THE SOUTHERN GENERAL ELECTRIC POWER COMPANY, LIMITED, BOSHYKEDDON. (Figs. 17 and 18.)

Duty—30,000 lb. steam per hour.
Vacuum—26.5 in. (barometer 30 in.).
Injection water—2,800 gallons per minute, temperature 65 deg. F.
Air pump capacity—54 lb. air per hour with atmospheric vacuum and water temperature.
Small motor to operate condenser about on loads over one-third and up to two-thirds full load.
Small ejector to start steam on loads of one-third full load and under.
Both ejectors to work on loads over two-thirds full load.

Time.	Load.				Vacuum.			Steam Pressure Ejectors.		Steam Pressure Air Pump.	Injection Water.		Extraction Pump.			
	Volts.	Amps.	P.F.	Kw.	Turbine.	Cylinder.		"G."	"I."		Inlet.	Outlet.	Press. Gauge.	Amps.	Volts.	H.P.M.
8.45 p.m.	6,350	180	0.76	1,970	29.1	26.3	26.5	180	Shut off.	122	—	—	5	52/60	440	480
11.0 a.m.	6,350	300	0.7	2,660	29.6	26.5	29.00	122	120	125	62	70.5	4	50	440	480
12.0 a.m.	6,350	300	0.7	2,183	29.6	26.5	29.00	122	125	125	62	70.5	4½	50	440	480
12.5 a.m.	6,400	270	0.7	2,000	—	26.5	29.00	Shut	125	125	62	70.5	4½	50	440	480
2.15 p.m.	6,400	265	0.76	1,760	29.5	26.25	29.50	125	120	120	62	72.5	5	62½	440	480
2.45 p.m.	6,450	265	0.76	1,970	29.5	26.5	29.50	125	127	127	62	72.5	5	62½	440	480
3.0 p.m.	6,500	280	0.74	1,980	29.5	26.1	29.6	125½	120	120	62	69.5	4	49½	440	480
3.30 p.m.	6,500	280	0.7	1,820	29.6	26.5	29.6	120	120	124	62	68.5	3	39	440	480



There is a central station near Glasgow where this type of apparatus is at work with a multijet condenser. Sometimes when changing machines there is liability partially to lose the water for a minute or so, but none of the staff ever have to trouble about the water for a minute or so, but none of the staff ever have to trouble about the ejectors, and as soon as the water comes back again the vacuum at once builds up, and the set is never shut down through failure of the air pumps. As a matter of fact, in the case above stated, it is highly probable that during the period that the water supply to the condenser is very small there is an air passage be-

tween the water spaces of the other condensers in the station and the multijet plant which would allow of a very excessive quantity of air getting into the condenser on load. This also shows that stability, the sixth claim, is another salient point.

That high vacuum can be obtained is proved by the fact that this apparatus is now being used in the French Navy and mercantile marine, as well as on some land installations for refrigerating purposes, and installations are at work where the maximum vacuum obtained is within 1 mm. of the barometer. For condensers the best results yet obtained by the French Westinghouse Company are within 5 mm. of the barometer.

From the foregoing remarks it would appear that this type of air pump is ideal for use on board ship, and particularly in the Navy, where space is so valuable and weight of such consideration, and to illustrate this point Fig. 14 has been produced, and represents to the same scale an Edwards air pump, Leblanc rotary air pump, and Leblanc multijector for a steam duty on a surface condenser of 40,000 pounds per hour, water at 60 degrees F., vacuum 28½ inches, and barometer 30 inches. The weights are approximately 20,832 pounds, 4,480 pounds, and 97 pounds, respectively. For land work it is equally suitable, and will soon supersede the rotary pump in many power stations. For sugar refineries, chemical, and other allied works it should prove exceptionally attractive, and take the place of many reciprocating dry-air pumps.

It may be advisable to point out here that this pump is purely a dry-air pump, so that for surface condensers an additional water or condensate pump is still required, and for jet plants the usual extraction pump.

Fig. 15 shows the air-dealing capacity of various sizes of multijector pumps taken from the actual tests. The maximum vacuum in each case is equivalent to the theoretical, the slight difference at the origin of the curves being due to the different test conditions.

The table of tests, H, is taken from a plant installed at the Scottish Central Power Station at Bonnybridge at a date six months after the plant was put on commercial load. This company has just decided to order another plant, and has specified Mirrlees-Leblanc multijector air pumps.

Fig. 16 shows the time taken to start up a Hick-Breguet *ejectair* and a Mirrlees-Leblanc multijector.

There is no doubt that considerable improvements have been effected in air-pump design during the last few years, nevertheless there is still room for further progress, and it is to be hoped that when the British engineer has time once again for research work we shall have to drop all our present-day notions of efficient air pumps for a type which will render all others obsolete.

"GARABED."

We had originally intended not to dignify the doings of Mr. Garabed T. K. Giragossian, of Boston, by discussing them on this page—for "dignify," we hope, is the right word. We have received so many inquiries from our readers, however, and governmental attention to Mr. Giragossian has reached such a point, that we feel constrained to reverse this decision.

This inventor claims to have discovered a way to utilize, without burning of fuel or other expenditure of labor or material, what he calls "free energy." He proposes that the Government investigate the matter, at his expense, through fully competent and properly accredited scientists; and he wishes, if their report be favorable, to assign to the United States the rights of use in his invention, to the end that we may employ it to bring about the end of the war.

Mr. Giragossian has received more attention than is usually accorded to those who bring forward schemes for ending the war at 7 o'clock tomorrow morning. In the first place, he came out of the intellectual metropolis of the universe armed with letters from three men who may well be described as prominent citizens of Boston, since one of them is director of music in the public schools there, another president of the board of trustees of the public library, and the third a mysterious combination of artist and financier and draftsman who has supplied the sinews of war to the good Armenian and made his working drawings for him without learning anything about his working details. These three gentlemen spoke so highly of Mr. Giragossian's "integrity and intelligence" that a committee of the House of Representatives found it necessary to refer twice in one brief paragraph to these sterling characteristics. And even before this, the inventor's flow of language had so captivated another chairman that he stated: "We were all impressed with his sincerity and honesty. * * * He did not tell us what he had, and he would not illustrate his invention to a subcommittee of five, but we were all impressed with his honesty and level-headedness."

In other words, Mr. Giragossian came to Washington, he saw, and, with his silver-tongued eloquence, he conquered. So runs the official version. But as a matter of fact, confronted with a proposition like this, there was really but one thing for Congress to do. No matter how absurd it seems on its face, no matter what its source, such an offer cannot be flouted. We are entitled to protect ourselves against a man who tries to sell us something; but when he offers to let us decide whether we want what he has, and, if we do, to give it to us—well, there can be no argument. We have nothing to lose and everything to gain, and we simply have to pay attention. That, indeed, is what we have done; at the present moment a joint resolution of the two bodies of Congress, enacting into law Mr. Giragossian's proposals, is somewhere along the legislative main line, and will presumably reach in due course the terminus at which the President wipes his pen and attaches his signature. Congress would have done well to have apologized less profusely for this action, for apology was not called for.

Let it be emphasized again and again, howeevr, that this in no way binds the United States to Mr. Giragossian, and, above all, that it in no way constitutes official endorsement or acceptance or approval of his claims. Indeed, what these are is absolutely not known, for the good and sufficient reason that Mr. Giragossian won't tell. In this he is hardly to blame; for if we had a valid invention which would do what he claims his will do, we should probably behave just about as he has behaved. But it is always given to discuss possibilities and probabilities; and we may therefore proceed to this phase of the matter.

The worthy Armenian, in what little he has said about his invention, as distinguished from the reams of stuff he has had to say about himself, has seemingly attempted to give the impression that he has hit upon some scheme for commercial utilization of the cosmic forces, for harnessing the power that makes the world go 'round and the heavenly bodies move in their paths. "Free energy" could mean this, and we fail to see what else it could mean. But until the protegé of Boston's distinguished public servants condescends to tell us, in intelligible terms, just what he proposes to do, we cannot discuss his scheme adequately. We can, however, point out that almost certainly he is on a par, as far as proper development of working details is concerned, with another Bostonian—or can it be that he himself is the one—who was anonymously reported to us several years ago as prepared to retard the earth in its rotation and impound the energy thus released?

Of course, human labor and the burning of fuel and the various other

performances which cost money are not a prerequisite to the manifestation of power. Gravity and atomic forces and electricity and magnetism and the tremendous energy of the sun's radiation act all the time, regardless of our will or our works. But when we try to direct one of these forces we have always heretofore had to do so by application of another force, put into play by our will and works in a way that does cost us labor and material. If Mr. Giragossian, with his nondescript collection of character witnesses, as lacking in scientific standing as their protegé, wants us to believe that he has found an escape from this stern necessity—well, he asks too much.—"Scientific American."

AN ILL-ADVISED REQUIREMENT.

One of the regulations designed to reduce the visibility of merchant vessels, which is now under discussion by the Ship Protection Committee of the Shipping Board, provides that all steam vessels sailing from the United States for the war zone must carry a sufficient supply of anthracite coal to carry the vessel for not less than two daylight periods, and that this coal must be used during the daylight runs while within the war zone. While the purpose of this regulation is commendable, the results sought are not likely to be accomplished if the rule is put into force. In the first place, an unlimited supply of anthracite coal is not available in many of the United States ports from which vessels are constantly sailing for the war zone, and in all probability difficulties will be experienced in furnishing an adequate supply for every vessel, especially in Southern ports, resulting in delays in sailings. Even if the supply of anthracite did not enter into the question there is a more serious disadvantage which must be faced, due to the fact that, on vessels with boilers operating on natural draft, anthracite cannot be burned at a sufficiently high rate to keep the vessel at her normal speed. On an American liner, for instance, the change from bituminous to anthracite coal reduced the speed of the vessel two knots. As speed is fully as essential as low visibility in escaping submarine attack, it would appear that whatever advantage would be gained in one direction would be offset by losses in another. Furthermore, the average fireroom crew lacks experience and skill in burning anthracite coal, and this factor will undoubtedly have an adverse effect on the steaming power of the vessel. In all probability better results would be obtained by more careful training of the firemen in the smokeless combustion of bituminous coal, for under careful supervision and with proper conditions in the furnaces the production of smoke can be greatly lessened and at the same time full speed maintained.—"International Marine Engineering."

THE LIBERTY MILL.

110-INCH PLATE MILL BUILT IN RECORD TIME BY CARNEGIE STEEL COMPANY FOR ROLLING SHIP PLATES FOR EMERGENCY FLEET.

The construction of the 110-inch Liberty Plate Mill by the Homestead Steel Works of the Carnegie Steel Company in six months is an engineering achievement which justly deserves mention, even at this present time of rapid construction of buildings for emergency uses.

The Liberty Mill is an emergency mill built in the first instance to roll ship plates for vessels under construction for the Emergency Fleet Corporation, but it has been built in the most substantial way to serve the nation when the war is done. A mill of this size had not been under consideration at Homestead Steel Works when the order to build the mill was received. There were no spare parts which could be taken from other

mills, no engines, no machinery available. No pencil had ever been put to paper in the way of preliminary estimates or designs.

WORK INVOLVED.

The construction of the mill required the removal of 70,000 tons of scrap and miscellaneous material on the site (2,400 carloads); 71,950 cubic yards of excavation; 50,120 cubic yards of back fill; 17,147 cubic yards of concrete; the laying, changing and removal of 40 miles of temporary track; the construction of a 5-foot sewer 1,000 feet long; it required the construction of an addition to the Carrie Furnaces power house with a new 46-inch by 66-inch Snow twin tandem gas engine direct connected to a new General Electric Company 3,750 KVA generator; the building of a transmission line two miles long across the Monongahela River on two towers 115 feet high and 1,007-foot span; the building of seven mill buildings with a total area of 215,000 square feet; the manufacture, fabrication and erection of 3,390 tons of structural steel; together with the heating furnaces, rolling mill, 4,000 horsepower main motor, subsidiary motors, straightening rolls, shears, cranes, etc., necessary for the operation of the mill itself.

LABOR.

The mill was built by men whose hands were already full; men whose works were operated to capacity with unusual difficulties at an unprecedented time; men whose minds were occupied with the problems of insufficient coal supply, scarce and inefficient labor and transportation difficulties. Ordinary labor was practically unobtainable; in consequence steam shovels and machine-tool methods were in use. Orders were received to build the mill on April 17, 1917, but it was six weeks after that date before it was possible to secure laborers.

Volunteer labor saved the day. Officials and clerks from various Homestead Steel Works offices, both general and departmental, officials and clerks from the general offices, Pittsburgh, Homestead Steel Works department heads, mechanics of all trades, skilled and semi-skilled mill workmen, were represented in the forces which on Saturday afternoon, Saturday night and Sunday cleared up the stagnation which had accumulated during the week on account of the shortage of regular laborers. At one time as many as 450 volunteer workmen were on the job.

ORGANIZATION.

The work was carried forward under the supervision of A. A. Corey, Jr., general superintendent, Homestead Steel Works; R. H. Watson, assistant general superintendent, and J. W. Grady, manager producing departments. Daily conferences were held at the site at 7:30 A. M., and each division of the organization had the direct oversight of that portion of the new plant which it was finally to operate, including design of machinery, its purchase, etc. F. B. Schaeffer, chief civil engineer, was responsible for the grading of the site, laying of foundations, sewers, underground work, equipment, back fill, railway tracks, etc. The design of the mill buildings and the general mechanical engineering work was in the hands of A. W. Soderberg, chief mechanical engineer. Cranes, motors, and all electrical equipment came under C. A. Menk, chief electrician. Richard Moon, master mechanic, had the oversight of the mill machinery, pumps, etc. Charles S. Frye was superintendent of construction, and Kenneth Lean superintendent of transportation and labor. James Horton, efficiency engineer, acted as special ambassador to the various contracting firms building mill equipment. Each department worked out its own problems.

SITE AND FOUNDATIONS.

The 110-inch plate mill and its tracks cover an area of nine acres on the Monongahela River, west of the Howard Axle Works in West Homestead. On this site on April 17, 1917, were 70,000 tons of scrap and miscellaneous material, or 2,400 carloads. Thirty thousand tons of scrap, 2,000 tons of loam, 700 tons of black clay and odd lots of sand, gravel, chrome ore, etc., had to be moved before the first pick could be sunk, and the first pick was a 73-ton 2½ cubic yard Marion steam shovel borrowed from the Union Railroad.

The foundations are all concrete, continuous for the brick buildings and of the single-pier type for the main mill building columns. The concrete was mixed in eight Smith & Co. concrete mixers, four of them of ½ cubic yard capacity and the other four the small mixerette, 5 cubic-foot size.

MILL BUILDINGS.

The group of buildings comprise seven distinct units at West Homestead in addition to the power-house extension at Carrie Furnaces. The slab-yard building is 70 by 293 feet; the heating-furnaces building, 106 by 293; the mill building, 70 by 293; the shipping building, 80 by 864; the shear building, 80 by 764. The mill superintendent's office, 46 by 64 feet, is of semi-fireproof construction and contains offices, vaults, toilet and rest-room.

It is customary in rolling mills to have the mill building itself contain equipment in the way of pumps, motors, accumulators, etc. In this installation all of the subsidiary machinery has been removed from the main mill building and segregated in a special mill office, comfort and power building, 242 feet long by 40 feet wide, built of brick and covered with Barrett specification roofing. This building contains three 500-kilowatt transformers built by the Pittsburgh Transformer Company, two 1,500-cubic foot Bury air compressors, two 5½- by 12-inch Epping-Carpenter duplex pressure pumps, two water filters, return-water tank, 24-inch accumulator, and in addition the physical testing laboratory, physical-test bending room, physical testing machine shop, locker and wash room, with 288 steel lockers, inspector's offices, janitor's room, oil room, toilet room, rest room, vault, etc., and has its own heating plant in the basement.

The separation of machinery from the mill proper has been carried a step farther so as to make the installation clean, clear and safe. The 4,000-horsepower motor which drives the mill is housed in its own separate brick building, built under the main mill roof and containing in a separate room the switches, resistance coils and other accessory electrical equipment.

The steel work for the mill buildings was designed by the American Bridge Company. It was fabricated in part by the Ambridge plant of that company and in part by the Upper Union Mills Fitting Shop of the Carnegie Steel Company. The steel work for the Carrie Furnaces power-house extension was detailed at Homestead Steel Works and fabricated in its own shop. The ability of the works to manufacture and fabricate its own steel and to do machine shop work was quite a factor in the rapid construction of the buildings.

Buildings and machinery were designed and ordered with any rechecking. Drawings came out of the drafting room daily, and the most remarkable feature of the engineering work was that there were absolutely no mistakes. Not one piece went out of the shops that had to come back.

The mill buildings were erected by the American Bridge Company with its own force. Work was begun on Thursday, August 2, and finished on September 22 in the record time of forty-nine days.

POWER.

This mill is the first electrically-driven plate mill to be operated by the Carnegie Steel Company. It is entirely electrically driven from start to finish. The main motor is a 4,000-horsepower General Electric motor. The crane motors, table motors, shear motors, motors for driving pumps, etc., were built by the Westinghouse Electric & Manufacturing Company.

The power furnished is alternating current, 3-phase, 25-cycle, 6,600-volt, developed at a central power station at Carrie Furnaces by a 46-inch bore by 66-inch stroke Snow twin-tandem gas engine directly connected to a General Electric 3,750 KVA generator. Power is transmitted to the mill on overhead lines carried on poles. This line spans the Monongahela River on two towers 115 feet high having 30-foot bases. The main mill motor takes current at the original voltage. At the subsidiary motors the voltage is reduced to 220.

ROLLING MILL MACHINERY.

The mill itself is a three-high plate mill with 110-inch length of roll body. Top and bottom rolls are 36 inches in diameter, middle roll 21 inches. The rolls are chilled iron and were made by the United Engineering & Foundry Company, of Pittsburgh.

The mill was designed by Homestead Steel Works and built jointly by Macintosh, Hemphill & Co. and the Mesta Machine Company, Pittsburgh. Mackintosh-Hemphill made the mill, large castings for charging machines and cranes, including mill housing. Machining was done by Homestead Steel Works. Mesta Machine Company made the pinions, the pinion housing, the pinion shoes and the mill shoes. The practice on motor-driven mills has been to use machine-cut pinions. It was impracticable to have these cut and so they were made machine molded and operate as noiselessly as any cut gears.

The mill is served by seven cranes and two overhead traveling charging machines, namely:

One 15-ton crane in slab yard.

One 50-ton crane in mill building.

Three 10-ton double-trolley cranes in shipping building.

Two 10-ton single-trolley cranes in shear building.

Two 10-ton overhead traveling charging machines in heating-furnaces building.

All these cranes and their equipment were made by Homestead Steel Works itself. There are three main shears, two side shears with 144-inch knife, and an end shear with 110-inch blade, all made by the Morgan Engineering Company, Alliance, Ohio. The two scrap shears were made by the Cleveland Punch & Shear Works. The straightening rolls were made by R. S. Newbold & Son Company, Norristown, Pa., and consist of a machine with nine rolls 14 inches in diameter and 116 inches long. The roll tables were made by Homestead Steel Works in their own shops.

The heating-furnaces building houses eight regenerative heating furnaces. At present furnaces are heated with natural gas. They may be heated with either natural gas, oil or tar. Ultimately, when the full schedule of improvements is completed, the method of heating will be by by-product gas from the Clairton coke ovens in conjunction with a small quantity of tar.

The mills rolls plates up to 100 inches wide and 3/16 inch up to 2 inches thick. It rolls from slabs, and sufficient steel for it is not available at present from the Homestead furnaces. The slabs come from Farrell Steel Works & Furnaces, Ohio Steel Works & Furnaces and Donora Steel Works & Furnaces, together with such miscellaneous slabs as can be taken from the present Homestead slabbing mills.

The best production to date is 266 tons, single turn, ten hours, with six furnaces. With an ample supply of steel and orders of a character suitable for maximum production, it will make 16,000 tons per month.

The cost of the mill will approximate $2,500,000 (£512,500), including therein the cost of the new gas engine, generator, etc., at Carrie Furnaces. On double turn it will employ approximately 250 men.

The average number of men employed under Homestead Steel Works organization in October, 1917, was 12,132, and the production of steel ingots was about 200,000 tons. This is not the maximum production for this plant. Tonnage has been reduced by reason of transportation difficulties, lack of coke, labor, etc.—"International Marine Engineering."

THE WAR'S EFFECT ON MERCHANT SHIPBUILDING.*

ENORMOUS TONNAGE REQUIRED BY THE GOVERNMENT—INCREASED SHIPBUILDING FACILITIES — TYPES OF SHIPS — STANDARDIZATION — LABOR PROBLEMS.

By Homer L. Ferguson.†

There have been two schools of thought regarding the proper method of meeting the German submarine campaign. One school has held that only by the production of the maximum number of ships in the shortest space of time until the production of new vessels should more than equal the destruction of vessels in being would the question be answered. The other school has held that there were vessels in plenty to feed England and to carry our troops to France and to feed them, and that the proper method to meet this menace was to build a large number of submarine chasers, destroyers and other vessels of war to go after the submarine itself.

The exponent in an official way of the theory of building the maximum number of ships is, of course, the United States Shipping Board, with the Emergency Fleet Corporation as the largest individual owner and builder of merchant ships anywhere, either at this time or at any other time, in the history of the world. All of the shipbuilders have had their new ships commandeered by the Shipping Board, and these ships are now being pushed to completion under the direction of the Shipping Board. In addition to the ships under construction at the time the Shipping Board took over all these vessels on August 3 last, the Shipping Board has placed contracts for 200 to 300 additional steel ships and for about 200 wooden and composite ships. The English have built ships at a very rapid rate during the past six months, but prior to that time their capacity had fallen off woefully. The figures of British shipbuilders are that in normal years their capacity in tons was about 2,000,000. In 1915 they built only 700,000 tons. In 1916 only 550,000 tons. In the first six months of 1917 they built 500,000 tons, and by the end of 1917 they will have built during this year almost 2,000,000 tons, or practically their normal tonnage.

CAPACITY OF AMERICAN SHIPYARDS.

In comparison, it is interesting to look at our record, which is mostly in the future. The largest tonnage ever built by us in any year prior to the present year was in 1908, when we built 615,000 tons. In 1917, or at the end of the fiscal year of 1916–17, on July 1, the total tonnage turned out in merchant ships in this country was close to 800,000 tons. Of course, we expect that that will be bettered very much, and the production, except for labor difficulties, would have been very much greater than that; but it is a far cry from 800,000 or 1,000,000 tons of ships a year to a prospective

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2,500,000 to 5,000,000 tons of ships a year, and especially when we consider that the best the first shipbuilding country of the world ever did in any one year was only 3,000,000 tons.

Now as to the methods of turning out such an enormous tonnage in a short time. If the war does nothing else for shipbuilding, it will at least demonstrate that the individual tastes of individual owners need not be met in every instance. Hitherto ships have not been built alike for the reason that ships were built by units and tens instead of by thousands. But with a single ownership, like the Emergency Fleet Corporation, it becomes possible for the owner to specify that all ships shall be alike, and thus we have standard ships!

There is an idea extant that if a ship is called "fabricated" or "standardized" it becomes rather easy to build. It does become easier to build, but it is not any easier to drive a rivet in a fabricated ship than in a common ship, and the 7,500-ton fabricated ship will have about 650,000 rivets and some riveters must do a considerable amount of work to drive those rivets in that ship the same as in any other ship. The most onerous work in building a ship is in the driving of the rivets; and if any concern is going to build one ship a week, it must drive 650,000 rivets a week. When we consider that the best rivet drive is by the Union Iron Works, where they drive 250,000 rivets a week, and the next best by the three largest shipyards on this coast is about 200,000 rivets each per week, the problem of driving 650,000 rivets a week assumes a rather difficult aspect to the casual observer. Still, we all hope it will be done, but if it is done it means the importation of an enormous number of new workers into the industry.

THE FABRICATED SHIP.

The fabricated ship has the center of the stage at the present time. In England a standardized ship was adopted which is very much like a fabricated ship, and they are being turned out at a rapid rate. The fabricated ship, like the English standard ship, has the disadvantage of low speed, and at the same time it has the advantage of a large carrying capacity. The records of submarine sinkings, such as have been gathered, indicate that speed is probably the safest defense against the submarine. The number of very slow ships sunk is very great in proportion to the number attacked. Probably 90 per cent. of the slow ships are sunk when attacked by a submarine. As the speed becomes greater the immunity of the ship from torpedo attack becomes greater—not only because she presents a more difficult target, but also because the vessel may run away from the submarine in darkness or in daylight if she has a greater speed than the submarine has on the surface, and also because the vessel may be maneuvered much more quickly. One of the most gratifying things about submarine warfare is that the torpedoes are not making the speed as in the early days of the war. The speed of the torpedo has undoubtedly become much decreased. Just why we do not know, although it is probably due to the absence of tungsten and manganese and some other ingredients now so difficult to obtain in Germany. Vessels of 16 to 17 knots have actually dodged the torpedoes.

The vessel of 16 or 17 knots is also needed to carry soldiers. We are shipping all our soldiers to Europe in vessels of 14 knots or over because it is thought to be too hazardous to ship them in slower vessels. We have at the present time in this country enough vessels in commission to ship soldiers to Europe at the rate of not more than 1,000,000 a year. That includes all vessels of 14 knots or over. But by building a large ship as well as a speedy ship an additional factor of safety is obtained, which may be illustrated by stating that a torpedo will blow just as big a hole in a little ship as in a big ship. When a small, short ship has a large hole in

it a relatively larger proportion of its length is open to the sea. When a similar hole is blown into a long ship a very much smaller proportion of that vessel is open to the sea. So that it is perfectly possible to design, and practical, indeed, to design, a ship of 500 to 600 feet long, so subdivided that it will take two and probably three torpedoes of the German variety and still stay afloat. The design, however, must be of such character that when the compartments are flooded the ship will still remain upright. The ordinary passenger ship is modeled with a small metacentric height, and three-fourths of them, when bilged or torpedoed, will probably turn over before sinking.

A great deal of talk was heard years ago, when the *Titanic* went down, about making ships safe. The *Titanic* was probably the safest ship that ever crossed the ocean, with the exception of the *Olympic*. The *Empress of Ireland*, the *Lusitania* and the whole host of passenger ships of which we have any record, have turned over as they went down, and they probably righted after they went down. So that in designing a ship that will be safe against submarine attack not only must the question of subdivision be taken care of so as to limit the length of the vessel that may be damaged by a torpedo, but also that the vessel when torpedoed will remain upright and be able to get into port. A number of vessels have been torpedoed and have been towed into port and repaired on the other side, even after the crews had deserted them.

The larger type of ship which will probably be built before the war is over by a number of concerns in a position to undertake this work will be designed with a high speed for troop transport, with a close subdivision to make the ship safe, and with a large amount of stability so that the vessel will also be safe against turning over.

TYPES OF VESSELS DEVELOPED.

The war so far has developed, or called attention to, three distinct types of vessels: The standardized or fabricated ship which both England and the United States have adopted, the large troop transport which has been described, and, in addition, a submarine chaser.

In addition to these three types of ships brought out by the war there has been a very definite attempt to rejuvenate what some of us have considered a dead industry. We have a few of the older wooden shipbuilders left, but the building of a wooden vessel, to my mind, is so much more difficult than the building of a steel vessel and requires so much higher skill than does the building of a steel vessel, that I wonder how they can never expect to build them.

WOODEN SHIPS INEFFECTUAL.

I do not believe that a 288-foot wooden ship built in a lumber yard, either South, West or North and necessarily by men who are not expert ship carpenters, will ever relieve any great distress on the other side of the water, caused by lack of fuel or food. On the other hand, these vessels may be used in our own trade at home. My chief regret is that they are not made shorter, because when such ships are over 200 feet long it is rather difficult to hold the ends up. If we get more vessels through building wooden ships, let us build them, but it is unfortunate that the type was settled before many professional wooden shipbuilders were on the job. The wooden shipbuilders, of course, have taken a good many men from the steel shipyards, and have lessened production to just that extent; but, on the other hand, they will add a very considerable tonnage, now that practically all our large coastwise ships are either going abroad or are in way of being commandeered for foreign service, so that the wooden ship will serve a useful purpose.

STANDARDIZATION.

The standard ship probably offers the best opportunity for a large increase in our tonnage. The standard ship has been built on the Great Lakes for years. Shipbuilding there has become largely a manufacturing operation, for the reason that the ships themselves simply form a part of the manufacturer's business of carrying iron ore from Lake Superior down through the canal to the eastern points, where it is shipped to the steel furnaces. The loading and unloading of the ore permitted the standardization of docks, hatches and so on, and the method of construction became standardized because the work which the ship did was standardized. Many ask why we do not build ships just as they do on the Lakes, and the answer is that we do not do the same work that they do. On the coast, with the coal trade, there has been some effort toward standardization. With the standardized ship the main benefit is not in the hull of the vessel, or in the fabrication of the vessel in the steel mill. The main benefit is that the machinery and the outfit, the winches, the pumps, the steering gear, the engines and boilers and all the thousand and one things that go into the ship are standardized and may be produced in large quantities at some point other than the yard of the builder. The standardization of the fittings of a vessel are more important to my mind than the standardization of the hull. The amount of money saved in the latter will be the money to build the templets, patterns, etc., amounting to not more than 12½ to 15 per cent. of the labor cost of the hull. We are told, too, that all the riveting will be done at the steel mills. It probably will not, and if 20 per cent. is done it will be more than I think will be done unless the railroads will carry pieces of steel as large as this auditorium.

STANDARDIZED SHIP FITTINGS.

Shipyards have naturally tried to standardize within their own practice the fittings that go on the various ships they build, and a great step is being made in that direction by one of the largest corporations in the country. The Bethlehem Company owns so many shipyards that they are in a position to standardize as between their various yards, with the result that there will be considerable saving in cost and in time in the production of all the things that go into a ship. But in building the standardized ship, or in building any other ship, the real problem is not the problem of equipment; it is not the problem of money even; it is not, except in part, the problem of location, but it is the problem of men. A very simple lesson in arithmetic will indicate that if 100,000 rivets are to be driven in one day and any one gang of riveters will drive 300, then it will take so many men and boys and helpers to get that drive.

SHIPYARD LABOR THE REAL PROBLEM.

It is quite desirable to get back to our arithmetic again when we have built so many ambitious high hopes of what we expect to do. For instance, on the Delaware and on the Chesapeake, which we on the Chesapeake are pleased to consider the two largest shipbuilding communities in the United States, there are at present about 33,000 men employed in the shipyards. There could be employed to advantage in these same yards at this time a total of 48,000 men, and within the next six months, or the next nine months at the latest, these same yards, barring the fabricated shipyards, could employ to advantage a total of 60,000 men. So the problem becomes arithmetical again. You have 33,000 men working. To get the maximum production out of the shipyards already in existence you will need to use 60,000 men. Where are the 28,000 extra trained shipbuilders to come from? Add to that the fabricated shipyards, which, on

the Delaware will probably require 20,000 men, and in the vicinity of New York an extra 10,000, and we have in this immediate district need for approximately 60,000 new mechanics in the shipbuilding business.

150,000 SHIPYARD MECHANICS NEEDED.

According to Admiral Capps, there will be needed in the whole United States in shipbuilding within the next six months an additional 150,000 men. Now, consider for a minute where these men must come from or how we are to get them. In the first place, most shipbuilding men or mechanics require both skill and experience. We can give them the experience fairly quickly, but we have not time to give them the skill. In other words, it is necessary that men be recruited from industries where they do work similar to the work of shipbuilding in order that they may become quickly adapted to the work of shipbuilding.

The problem of oversea transportation absolutely has to be met before we can get into the war effectively, and the problem of sea transportation is probably the most difficult of all problems we have to face. Therefore, the men should be furnished for the solution of this problem, no matter what industry they are taken out of. The problem of building these ships becomes the most important single problem that confronts us at the present time.

CONTROL OF SHIPYARD LABOR.

I do not say that I favor the conscription of laboring men, but I do say this—that sooner or later the United States will tell all men who are essential in any industry or any occupation in the prosecution of the war that *this* is their job and they are to stay by it until the war is over. It is perfectly ridiculous that our men who are already in Europe and those who are going there now should for a minute suffer on account of some difficulties that arise between employers and employees in the great shipyards of this country. Not only should the man who pounds iron and drives rivets be told that he shall be kept on his job, but also every employee in the company, from the president down to the office boy, is to do the same thing, and he is not to seek or obtain other employment, except for good reason given the Government authorities. If we are in the war to win, no one should be timid about telling the laboring man or the unions or anybody representing them just what part they have got to play along with the part that other people should also play.

It is held by the National Council of Defense, and by a number of exalted bodies that the *status quo* as between employer and employee should remain the same during this war. Of course it should remain the same, and we all know that it is not remaining the same, and we all know that the labor leaders as a rule are unable to control the men. I do not see why of all the men who have been passed into the national army we should not have in as essential an industry as shipbuilding those of them who are mechanics on conditions set by the Government, and on pay to be approved by the Government, to work in these shipyards until the war is over, or until they should perchance prefer to go to the front instead of staying in the shipyards.

I understand that in Philadelphia it is very difficult to have any of the shipyard employees excused from military service. Of course, they have their own rules to go by in Washington, and we must all abide by them, but it is rather unfortunate that they should be differently interpreted in different parts of the country, and it seems to me most unfortunate that the shipbuilders have been allowed to go to the front. The experience of England was that they had to take out of the trenches not only the shipyard workers, but the ammunition workers, and send them back to the

factories again. It is a pity that we could not profit by that. That would be a solution that would help very much with this laboring-man difficulty.

FIXING WAGES.

The demand is largely in excess of the supply of men. The sympathy of the Government is with the men. At our navy yard under the law we pay the going rate in that vicinity and arbitrarily fix the rate at a larger rate than the going rate in the vicinity which we have to meet, and we are told in case of difficulty we must keep going, so it is only a question that can be worked out by the Government's representatives and the shipyard employers together.

I would be perfectly willing to see all the shipbuilders go down to the Navy Department and Shipping Board and say, "We will agree with you on these conditions and stand by them and will shut up our shop before we pay any more." Of course it can be said that a man who works is entitled to any wage he can get. Perhaps he is, but he is not entitled to stop working now, and he is not entitled to say that any other man shall not serve an apprenticeship now. He is not entitled to say that a helper shall not do a mechanic's work if he can do it. How perfectly ridiculous when a million and a half of the best young men that we have in the whole United States are serving an apprenticeship in the war, leaving their homes and going out to learn the art of soldiering in a very short time, and quite as difficult work to learn as riveting. What a ridiculous thing to say at a time like this that a man cannot get a job unless he has served his time at the trade, and tie the hands of the United States in this great war, which it must win if you and I are going to keep on being proud of living in the United States.

I was shown today in the case of a number of workers that the money which they could earn in a day was limited by the organization to which they belonged, and that if they earned that much money at 2 or 3 o'clock in the afternoon they would work no more, notwithstanding the fact that the management wanted them to do it. When will people ever learn that production by the use of labor-saving machinery to get a large production per man is all that gives us more than we ever would have otherwise or than we ever would have had before the days of large production? And yet they have limited production with the idea of giving more jobs to more men in time of war when we have not got enough men and have about three jobs for one man!

HOUSING SHIPYARD WORKMEN.

The condition of laboring men who are gradually coming in from the West to all the seaboard towns is a very difficult one, and a great deal of the basis of discontent is a lack of suitable housing conditions. That is ordinarily taken care of in the towns and cities by real estate people and those who build for investment, but at the present time that method is wholly inadequate, and this question will have to be considered along with the same program as that for our army which is being installed in cantonments all over the United States. The Government is waking up to this in the case of some recent contracts let for destroyers. The Government itself will finance the building of barracks or temporary hotels for the men so that several hundred can be housed at or near the shipyards where the work is to be done. One of the biggest problems England had was the formation of new communities and the building of whole towns. In some cases they were almost ten miles square, and in them were provided not only ordinary houses and living accommodations, but public parks, playgrounds, theaters and everything needed in a modern town.

The shipbuilding problem comes back to the question of men and willing men—willing to do work under conditions which their own Government and our representatives say are fair and honorable both to the Government and to the men, and I say that if they will not work under conditions such as the Government specifies and at the rates of pay which the Government calls fair, then they should either be made to, or made to let alone the men who are willing to do so.

I cannot imagine the United States letting stand in its way any band of men whatever when the national honor and national life, in fact, depends on our winning this war, and when we are sending forth our young men and boys by the thousands to France.

TREAT EVERYBODY ALIKE.

As a business man, I do not know of any man in the shipbuilding business who would care, and I, myself, do not care, how much of my income the Government takes, as long as it treats everybody else the same, and as long as the business is left with enough money to carry on its operations and pay a most moderate return to the owner and be able to serve the Government in the best way. I do not care how much of the profit of the business they take, and I will go further than that. If it becomes necessary for the raising of money, a conscription of capital would not hurt my feelings at all. We are willing, if necessary, to have the plants commandeered by the Government, under the terms of the United States and under conditions which they see fit to impose, but we do insist that managers, foremen, superintendents and men shall all be treated alike.

Those of us who are building ships or destroyers have a very difficult time in securing priority in getting material. We have received from the Navy Department and from the War Department and from the Shipping Board orders to proceed with the utmost vigor and despatch to do their work, each in the same tone of voice, each with equal authority, all vested in these various departments by Acts of Congress, so that actually what has happened is that the president of our company has had to settle the question of priority himself—an entirely improper function for him. The question of priority and preference is now the biggest industrial question there is, for the very simple reason that the total production of the United States has been over-absorbed by these war activities.

PRIORITY IN GETTING MATERIAL.

With the prices settled, the only important thing is, who gets his first, who second and who third. We have in the War Department six bureaus buying frequently in competition with each other. We have in the Navy Department five or six bureaus buying, or recommending to buy, not in competition with each other very much, but somewhat. In the Shipping Board we have a tremendous amount of buying going along more or less in competition with the War Department and the Navy Department. The Government simply wants to get the material in such quantities as it can, but it is simply impossible to satisfy all these bureaus and departments. What is the answer?

MUNITIONS MINISTER NEEDED.

The answer that has been found in Great Britain in the very first months of the war was to divorce the question of business, buying and price fixing from the naval and military activities of the country. A munitions minister was appointed; just the kind that we need. The Chamber of Commerce of the United States recommended that a board be appointed. I am a director in that Chamber, but I think a minister should be appointed,

because you want a *man* who will coördinate the interests of the country What have we in our country? We have great army cantonments. We have great army and navy orders from the Government and are competing day after day; we do not know who is first, second or third. I am sure that the Secretary of the Navy and the Secretary of War have both plenty to do to run the operations of their great departments without working a large part of the time with the question of buying, and whether they are going to get this first, second or third. We have a Priority Committee, but they cannot settle it, but only recommend it. Actually, the best buying is done by the official who is the most zealous, so that we run great risk, if that keeps on, of having our soldiers on the other side with plenty of guns and maybe gunpowder, but not enough baked beans or peas or something that they need. It must be done in some way so as to least upset the country.

Another reason that business should go on as well as possible because we have to earn some money somehow in order to meet the war taxes, and there is no need of disrupting it unnecessarily in buying in various ways. That has had a bad effect on the shipbuilding business.

FORGING SITUATION A WEAK POINT.

The weakest point in the shipbuilding business is the forging situation, in my judgment. There are not enough forges in the United States to turn out all the forgings required for the shipbuilding. The great Bethlehem Company, Midvale, and the Allis-Chalmers are practically the only large ones we have, whereas Great Britain has twelve or fifteen very respectable forging concerns. It is absolutely necessary that someone determine whether this or that or the other kind of forgings shall be first; otherwise a lot of us are apt to end up with hulls but no machinery in them. But we would very much appreciate it if we could go to Washington and say to someone that this is the most important, and that is the next, and that is the next. Instead of that, we appeal from one department to another and frequently end up with nothing.

I am mentioning this to you because it is very much in my heart and mind; I see it in Washington every time I go there, and I hope very much that next winter when the question comes up in Congress they will see fit to centralize the buying of war materials instead of scattering the work among he Navy Deparment and the War Department and the Shipping Board.

TRAINING SHIPYARD WORKERS.

We all know that it is a joy to have a few good mechanics and give them a job and forget about it, but we have to get away from that and to teach new people, which may, fortunately, be done with ships of standard make and duplicate makes. The biggest problem that the employment managers have at the present time is to get hold of the best material possible, house it as decently as we can, and teach it shipbuilding as quickly as possible. The problem almost dazes one to contemplate. It can only be solved by bringing in enormous quantities of new men, and which must be done without any hindrance. The leaders of the unions have stated that they would allow this to be done. We have to do it, and no matter what else happens we must insist upon the right to break in any number of new men in the business. The Shipping Board will back it up, and I am sure that sooner or later the Administration itself will insist upon that being done on such a comprehensive scale as to make it possible for the United States to carry out its great shipbuilding program, which *must* be carried out.

THE 170-HORSEPOWER MERCEDES AVIATION MOTOR.

By E. H. Sherbondy.

This motor is one of the five types standardized by the German War Office and is in very general use at the present time. Its maximum power is 176 H.P., at 1,450 r.p.m. This motor is of the fixed vertical type, has six cylinders in line, which are not off-set and is water cooled. The camshaft is carried in a special bronze case, seated on the immediate top of the cylinders, which in turn is driven by bevel gearing, a vertical shaft being interposed between the crankshaft and camshaft.

The water pump is located on this vertical shaft and serves to steady the motion of this shaft. Extending immediately below the camshaft is another vertical shaft, driven by bevel gears off from the crankshaft master bevel, terminating in a worm, which drives the multiple-piston oil pumps.

CYLINDERS.

The cylinders are made from steel forgings, as are the valve-chamber elbows, which are machined all over and are welded together. A jacket of light steel is welded over the valve elbows and attached to a flange on the cylinder, allowing for a water-cooling space, having a section of about 10 m/m or 7/16 of an inch. The bore of the cylinders is 140 m/m or 5½ inches, the stroke is 160m/m or 6-1/16 inches.

The volume of the compression chamber is .645 litres, the cylinder capacity being 2 litres .463. The ratio volumes is 4.818. The cylinders are locked to the crank case by means of dogs and long through bolts, which have shoulders near their lower ends and are bolted to the lower half of the crank chamber. All this gives an exceedingly stiff and light structure, and the construction has been imitated by a great many other builders.

There are a number of peculiarities in connection with these cylinders which are well worth noting. In Fig. 2, which is a longitudinal section through the motor, the water-cooling system will be clearly apparent. The water after leaving the water pump enters the top of the front cylinders and passes successively through each of the six cylinders. There are short tubes welded to the top of the water jackets which serve as a communicating gallery. This same system of cooling was used for years by the Panhard Company of Paris on their motor car engines. It has more recently been used in connection with the English White and Poppe Motor Truck Engines. Excellent circulation is had around the valve chambers and cylinder heads.

It will also be noted that a lug has been forged with the valve elbow, which serves as the anchor and steadiment for the camshaft housing.

PISTON AND CONNECTING RODS.

The pistons are built up of two pieces. First—A dropped forged-steel head, extending from which are the piston-pin bosses, and secondly, a cast-iron skirt, into which this head is screwed. The skirt carries three rings at its upper end and one on its lower end. There are also a couple of lubricating-oil grooves cut into this skirt.

After the head has been screwed into the skirt it is retained by two small rivets and is welded at a couple of points. The coefficient of friction between cast iron and steel is considerably lower than between two steel members, and there is less tendency for a cast-iron piston to cut or score than where a steel piston is used. A great many makers have tried steel pistons and afterwards given them up and gone over to cast iron. For instance, in the original Gnome engines, steel pistons with a single

brass ring were used. These were afterwards given up, and in all the Gnome engines now being built cast-iron pistons, with a percentage of steel in the mixture, are the invariable standard. The LeRhone rotating engine, which has necessarily steel cylinders, has a cast-iron liner forced into these cylinders for wearing surface.

There are also some distinct advantages possessed by this piston design. The skirt can be readily machined all over, and may have uniform wall

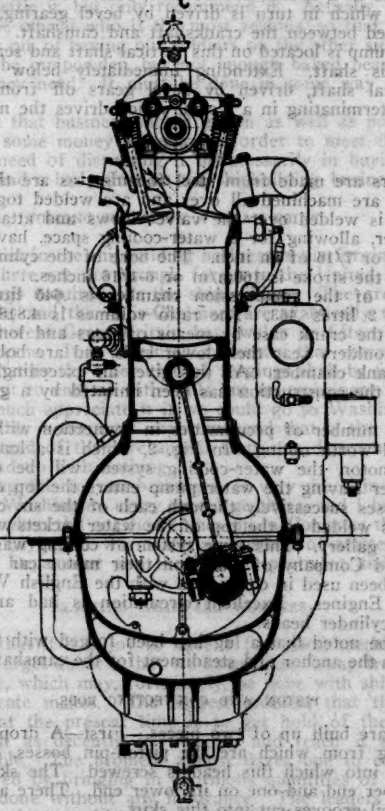

FIG. 1.—CROSS-SECTION VIEW OF THE 170-H.P. MERCEDES MOTOR.

thickness, which in itself greatly assists in preventing distortion. The piston pin is formed from a Chrome nickel-steel tube and is retained in the piston by means of a simple set screw and cotter pin.

The connecting rods are very short and very stiff, of I-beam section and are machined all over. At their upper end there is a floating bronze bush, which bears on the piston pin. Banded to the web of the rod is an oil tube for conveying lubricant from the crank pin to the piston pin.

THE CRANKSHAFT AND ITS MOUNTING.

The crankshaft is also a Chrome nickel-steel member and is carried on seven bearings. There are holes through each of the crank pins and main bearings fully half the diameter of the shaft, which are plugged by

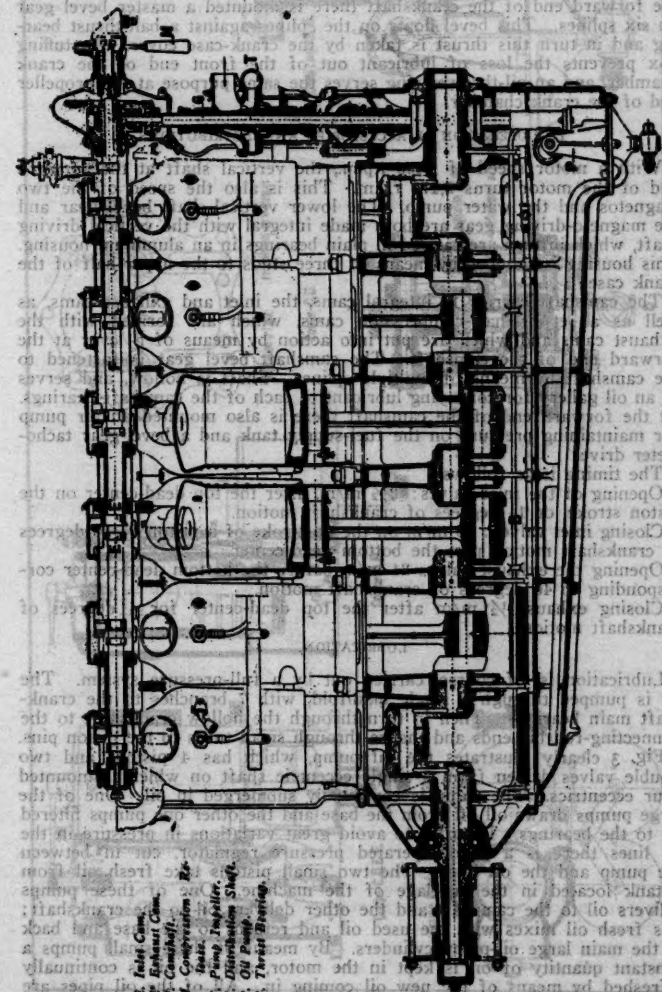

FIG. 2.—SECTIONAL VIEW OF THE 170-HORSEPOWER MERCEDES AVIATION MOTOR.

pressed-brass members. Small holes are drilled through the crank cheeks, which serve to convey lubricant from the main bearings to the crank pins. The propeller thrust is taken by a simple ball-thrust bearing at the propeller end of the crankshaft. This thrust bearing is seated in a steel retainer which is clamped between the two halves of the crank case. At the forward end of the crankshaft there is mounted a master bevel gear on six splines. This bevel floats on the splines against a ball-thrust bearing and in turn this thrust is taken by the crank-case cover. A stuffing box prevents the loss of lubricant out of the front end of the crank chamber, and an oil-thrower ring serves the same purpose at the propeller end of the crank chamber.

THE OPERATION OF ACCESSORIES AND DISTRIBUTIONS.

With a motor speed of 1,450 r.p.m., the vertical shaft at the forward end of the motor turns 2,175 r.p.m. This is also the speed of the two magnetos and the water pump. The lower vertical-shaft bevel gear and the magneto-driving gear are both made integral with the vertical driving shaft, which in turn are carried in plain bearings in an aluminum housing. This housing is clamped by means of three studs to the upper half of the crank case.

The camshaft carries 18 integral cams, the inlet and exhaust cams, as well as a set of half-compression cams, which are formed with the exhaust cams and which are put into action by means of a lever at the forward end of the camshaft. The camshaft bevel gear is fastened to the camshaft by means of multiple splines. Shaft is hollow, and serves as an oil gallery for conveying lubricant to each of the camshaft bearings. At the forward end of the camshaft there is also mounted an air pump for maintaining pressure on the fuel-supply tank and a bevel-gear tachometer drive.

The timing is as follows:

Opening of the inlet valves: 2½ m/m. after the top dead-center on the piston stroke, or 14 degrees of crankshaft motion.

Closing inlet valves: 15 m/m on the up-stroke of a piston or 36 degrees of crankshaft motion past the bottom dead-center.

Opening the exhaust valves 24 m/m before the bottom dead-center corresponding to 46 degrees of crankshaft motion.

Closing exhaust ½ m/m after the top dead-center for 4 degrees of crankshaft motion.

LUBRICATION.

Lubrication is, of course, carried out by a full-pressure system. The oil is pumped through a single manifold, with 7 branches to the crankshaft main bearings. Then in turn through the hollow crankshaft to the connecting-rod big ends and thence through small tubes to the piston pins.

Fig. 3 clearly illustrates the oil pump, which has 4 pistons and two double valves driven from a single eccentric shaft on which is mounted four eccentrics. The pump is continually submerged in oil. One of the large pumps draws oil off from the base and the other one pumps filtered oil to the bearings. In order to avoid great variations in pressure in the oil lines there is a piston-operated pressure regulator, cut in between the pump and the oil lines. The two small pistons take fresh oil from a tank located in the fuselage of the machine. One of these pumps delivers oil to the camshaft and the other delivers oil to the crankshaft; this fresh oil mixes with the used oil and returns to the base and back to the main large oil-pump cylinders. By means of these small pumps a constant quantity of oil is kept in the motor, and the oil is continually refreshed by means of the new oil coming in. All of the oil pipes are fastened into the lower half of the crank chamber very securely.

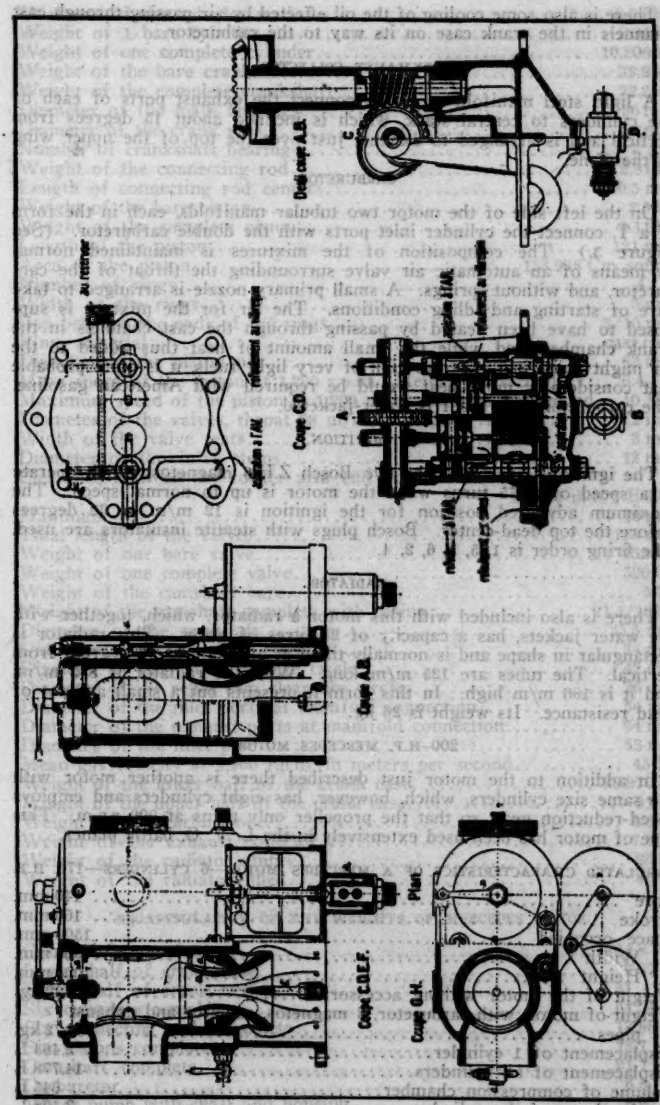

Fig. 3.—Details of Carburetor and Oil Pump, Mercedes Motor.

There is also some cooling of the oil effected by air passing through cast channels in the crank case on its way to the carburetor.

EXHAUST COLLECTOR.

A light steel manifold serves to connect the exhaust ports of each of the cylinders to central stack, which is inclined about 15 degrees from vertical and is arranged to exhaust just over the top of the upper wing of the plane.

CARBURETOR.

On the left side of the motor two tubular manifolds, each in the form of a T, connect the cylinder inlet ports with the double carburetor. (See Figure 3.) The composition of the mixtures is maintained normal by means of an automatic air valve surrounding the throat of the carburetor, and without springs. A small primary nozzle is arranged to take care of starting and idling conditions. The air for the mixture is supposed to have been heated by passing through the cast channels in the crank chamber, and while the small amount of heat thus added to the air might assist in the carburetion of very light fuels, it is quite probable that considerably more heat would be required with American gasoline. The carburetor body itself is water-jacketed.

IGNITION.

The ignition is furnished by two Bosch Z.H.6 magnetos, which operate at a speed of 2,175 turns when the motor is up to normal speed. The maximum advanced position for the ignition is 12 m/m or 32 degrees before the top dead-center. Bosch plugs with steatite insulators are used. The firing order is 1, 5, 3, 6, 2, 4.

RADIATOR.

There is also included with this motor a radiator, which, together with the water jackets, has a capacity of 25 litres of water. This radiator is rectangular in shape and is normally tilted at an angle of 30 degrees from vertical. The tubes are 125 m/m long. With the radiator is 835 m/m, and it is 186 m/m high. In this form it presents but a small amount of head resistance. Its weight is 26 kg.

200-H.P. MERCEDES MOTOR.

In addition to the motor just described there is another motor with the same size cylinders, which, however, has eight cylinders and employs speed-reduction gear, so that the propeller only turns at 900 r.p.m. This type of motor has been used extensively in the L. V. G. battle planes.

TABULATED CHARACTERISTICS OF A MERCEDES MOTOR—6 CYLINDERS—170 H.P.

Bore	140 mm.
Stroke	160 mm.
Space occupied	1500 mm.
Width	500 mm.
Height	1005 mm.
Weight of the motor without accessories	273.22 kg.
Weight of motor, with carburetor, 3 magnetos, radiator and exhaust pipes	306.72 kg.
Displacement of 1 cylinder	2.463 L.
Displacement of all cylinders	14.778 L.
Volume of compression chamber	645 L.
Total volume of the cylinder	3.108 L.
The compression pressure	4.818 kg. per Cm^2

Thickness of the cylinder wall	3.5 mm.
Weight of 1 bare cylinder	8.34 kg.
Weight of one complete cylinder	10.206 kg.
Weight of the bare crankshaft	32.2 kg.
Weight of the complete crankshaft	32.2 kg.
Diameter of the main bearings	56 mm.
Diameter of the crank pins	54 mm.
Number of crankshaft bearings	7
Weight of the connecting rod	2.31 kg.
Length of connecting rod centers	290.5 mm.
Weight of the bare piston	2.5 kg.
Weight of the complete piston	3.11 kg.
Length of the piston	121 mm.
Area of the piston	153.938 Cm. square
Number of rings	4
Width of the rings	5 mm.
Gap between the ends of the rings	1/10 mm.
Ring material	cast iron.
Diameter in the piston pin	32 mm.
Mean speed of the piston at 1,200 r.p.m. Meters per second	6.4
Maximum speed of the piston at 1,200 r.p.m. Meters per second	10 M
Diameter of the valves, throat 68 m/m; outside	72 mm.
Width of the valve seats	3 mm
Diameter of the valve stems	12 mm.
Clearance between the rocker arm end and the valve stem,	1-4/10 mm.; E 3/10
Maximum lift	10 mm.
Total length of the valve	153 mm.
Weight of one bare valve	452 Gr.
Weight of one complete valve	520 Gr.
Weight of the camshaft bare	5 kg.
Weight of the camshaft complete with casing	21.1/10 kg.
Diameter of the camshaft	25.5 mm.
Diameter of the cam base circle	34 mm.
Diameter of the cam clearance circle	42 mm.
Diameter of the relief cam clearance circle	36 mm.
Diameter of the inlet ports at manifold connection	54 mm.
Diameter of the exhaust ports at manifold connection	54 mm.
Diameter of the inlet pipe	53 mm.
Mean gas velocity at 1,200 r.p.m. in meters per second	45 M
Weight of the lower half of the crank case	39.970 kg.
Weight of the upper half of the crank case	31 kg.
Weight of the carburetor	6.6 kg.
Weight of the exhaust collector	7.5 kg.
Weight of the radiator empty	26 kg.
Weight of the radiator filled	40 kg.

RECAPITULATION OF THE WEIGHTS OF MERCEDES MOTOR.

Crankshaft complete	34.8 kg.
Lower half of crank case	39.97 kg.
Upper half of crank case	31. kg.
Six cylinders complete	61.24 kg.
Six connecting rods complete	13.86 kg.
Six pistons complete	18.7 kg.
Camshaft complete	21.1 kg.
Carburetor	6.6 kg.
Water pump with shaft and housing	6.6 kg.
Oil pump with shaft	5.85 kg.

Two operating magnetos	13.	kg.
One starting magneto	4.	kg.
Inlet pipe and ignition wire casing	11.	kg.
Miscellaneous parts	5.5	kg.
Total	273.22	kg.
Exhaust pipe complete	7.5	kg.
Radiator empty	26.	kg.
Total	306.72	kg.

THE 230 H.P. BENZ AERO ENGINE.

The following detailed description of the 230 H.P. Benz engines is based on information courteously placed at our disposal by officials of the British Air Board. Being of a far more detailed nature than any particulars that have hitherto been published relating to German aero engines, this description will be read with interest by all concerned in aero engine design and construction.

LEADING FEATURES OF THE ENGINE.

Following the usual German aero engine practice, the 230 H.P. Benz is of the six-cylinder vertical water-cooled type. Each separate cylinder is bolted to the crankcase by long bolts and studs, which pass through the crank-chamber top half and secure the crankshaft bearings between the top and bottom halves of the crank chamber.

Two inlet and two exhaust valves are fitted in the head of each cylinder. The valves are operated by overhead valve rockers working on ball bearings and by push rods on either side of the cylinders. The two camshafts, which run on plain bearings, are neatly arranged inside the top half of the crankcase, and the floating exhaust camshaft is provided with half-compression cams.

The pistons are of cast iron, fitted with three exceptionally wide rings, and the piston heads, following usual Benz practice, are supported by conical steel forgings riveted and welded to the piston crown, which bear on the center portion of the gudgeon pins, through slots cut in the connecting-rod small ends.

As in the 160 H.P. Benz engines, two separate two-jet carburetors are fitted, each having their air intake passages through the top half of the crankcase casting. Each carburetor supplies three cylinders through an independent branched induction pipe, built up of light aluminum tube.

The lubrication of the crankshaft and connecting-rod bearings is effected by a very neatly designed gear pump working in an auxiliary-oil reservoir formed in the bottom of the air-cooled base chamber.

An oil-sealed petrol pump of interesting design supplies petrol to the carburetors in conjunction with a supplementary pressure reservoir enclosed in the main petrol tank. The petrol pump is driven off the rear end of the inlet camshaft. The same driving spindle also operates the machine-gun interrupter gear and the tachometer drive.

CONSTRUCTIONAL DETAILS.

Cylinders.—Except for the steel water jackets, the cylinders are made entirely of cast iron. The water-jacketed heads, including the twin inlet- and exhaust-valve passages, are cast integral with the cylinder castings.

The bore of the cylinders is 145 mm. and the thickness of the cylinder walls tapers from 5.5 mm. at the base flanges to 6.5 mm. at the top of the cylinder barrels. The water jackets are of die-pressed sheet steel, built up from half sections and welded both at the vertical joints and at the

bottom of the water jackets to flanges machined on the outside of the cylinder barrels. The water jackets are exceptionally long, extending to within 45 mm. of the cylinder-base flanges. Seven annular corrugations are formed in the water jackets to allow for expansion, and three transversely in the sheet-steel crown of the water jackets.

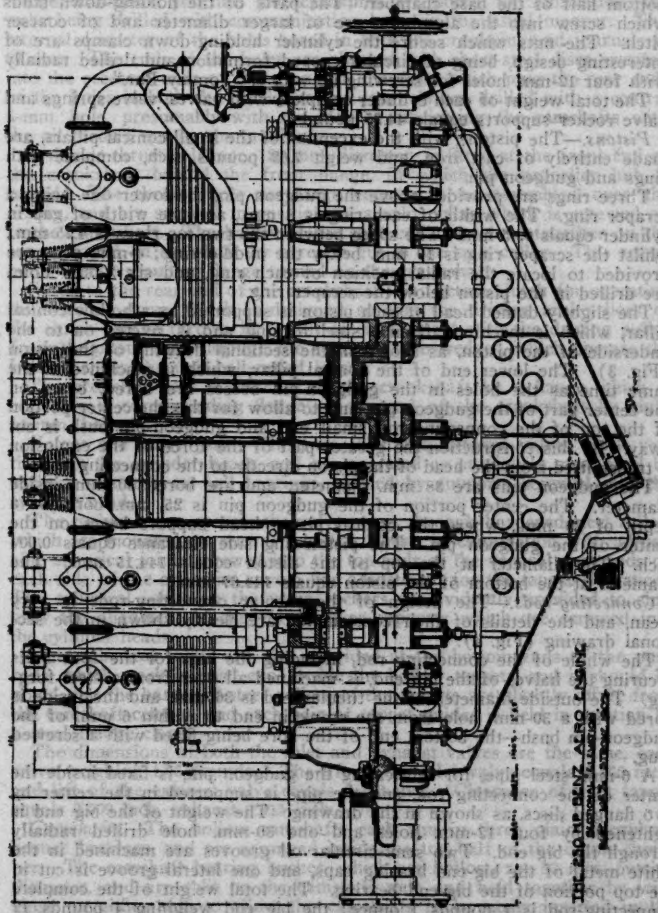

FIG. 1. SECTIONAL GENERAL ARRANGEMENT.

The water spaces formed in the cylinder-heads and the cooling of the valve pockets is well carried out. Dished plates are welded in position in the water space above the crown of each cylinder to deflect the flow of water on to the exhaust valve pockets. The diameter of the cylinder-head water connections is 60 mm.

The cylinder registers extend to 10 mm. below the base flanges into the crank chamber, and are held down by four 12-mm. studs, and also by dogs at four points. The dogs are secured by long studs which pass through the top half of the crank chamber and are screwed into the bottom halves of the main-bearing housings, which are cast integral with the bottom half of the base chamber. The parts of the holding-down studs which screw into the aluminum are of larger diameter and of coarser pitch. The nuts which secure the cylinder holding-down clamps are of interesting design, being of circular cupped formation and drilled radially with four 12-mm. holes for screwing up with a "Tommy Bar."

The total weight of each cylinder complete with valves, valve springs and valve rocker supports equals 44.25 pounds.

Pistons.—The pistons, with the exception of the small conical pillars, are made entirely of cast iron, and weigh 7.62 pounds each, complete with rings and gudgeon pin.

Three rings are provided above the gudgeon pin, the lower one being a scraper ring. The width of each ring is 8 mm., and the width of gap in cylinder equals 0.45 mm. The space between the two top rings is 4.25 mm., whilst the scraper ring is 10 mm. below the middle ring; 4-mm. pegs are provided to locate the radial position of each ring, and six 2-mm. holes are drilled in the piston below the scraper ring.

The slightly domed head of each piston is supported by a hollow conical pillar, which is machined from a steel forging and is riveted on to the underside of the piston, as shown in the sectional drawing of the piston (Fig. 3). The lower end of the conical pillar, which is machined at the same time as the holes in the gudgeon-pin bosses are bored, bears on the center part of the gudgeon pin, and to allow for this the center portion of the top of the connecting-rod small end and gudgeon-pin bush is cut away. By this construction the greater part of the force of the explosion is transmitted from the head of the piston directly to the connecting rod.

The gudgeon pins are 38 mm. diameter, and are bored 30 mm. inside diameter. The center portion of the gudgeon pin is 25 mm. bore for a depth of 20 mm., where the conical piston head support bears on the center of the gudgeon pin. The piston-ring side clearance equals 0.004 inch. The diameter at the top of the piston equals 144.15 mm. The diameter at the bottom of the piston equals 144.67 mm.

Connecting-rods.—The design of the tubular connecting-rods is very clean, and the details of their construction are clearly shown in the sectional drawing (Fig. 4).

The whole of the connecting rod, including the lugs for the four bolts securing the halves of the big end, is machined all over from a steel forging. The outside diameter of the tubular rod is 36 mm. and the inside is bored with a 30-mm. hole from the crankpin end to within 6 mm. of the gudgeon-pin bush—the bottom end of the bore being fitted with a screwed plug.

A 6-mm. steel pipe, for lubricating the gudgeon pin, is fixed inside the center of the connecting rod, and the pipe is supported in the center by two flanged discs, as shown in the drawing. The weight of the big end is lightened by four 12-mm. holes and one 30-mm. hole drilled radially through the big end. Two semi-circular oil grooves are machined in the white metal of the big-end bearing caps, and one lateral groove is cut in the top portion of the big-end bearing. The total weight of the complete connecting-rod is 7 pounds 1 ounce; the big end weighing 4 pounds 12 ounces and the small end 2 pounds 5 ounces.

Total big end side clearance equals 0.15 mm., and the float of the small-end bush between the gudgeon-pin bosses equals 14 mm.

Crankshaft.—The six-throw crankshaft runs on seven plain bearings, and weighs 109.25 pounds, including the propeller boss.

The cranks are, of course, set at 120 degrees, and the diameter of all the journals is 62 mm., whilst that of the crankpins is 60 mm. The length of the front journal bearing (propeller end) equals 79 mm., and the length of the other journal bearings equals 54 mm., with the exception of the rear end bearing, which is 55 mm.

The crankshaft journals and crankpins are bored for lubrication, the webs being drilled with communicating holes in the usual manner. The internal diameter of the holes bored in both the crankpins and journals is 27 mm. The ends of the holes are plugged with sheet-steel discs sweated into the recessed ends of holes, and all the discs which plug the rear ends of the holes in both the crankpins and journals are drilled with a central 5-mm. hole, presumably with the idea of lubricating the camshafts with the oil thrown out by the cranks.

A double-thrust ball race, 120 mm. diameter, is fitted at the front end of the crankshaft behind the front flange, to which the propeller hub is bolted. The thrust races are large enough to be assembled over the cranks, and are secured in position by a split collar which is screwed on to the crankshaft. The halves of the split collar are held together on the screw thread on the shaft by a recess cut in the split collar. This recess holds a corresponding flange turned on the crankshaft.

Fitted to the rear end of the crankshaft is a friction clutch for operating the wireless drive, which is designed so that the driving brake-shoes of the clutch can be thrown in or out of engagement with the driving pulley from the pilot's seat, through the action of two wedges, which operate the friction shoes through ball-ended levers.

The propeller boss is attached to the crankshaft by a flange which is bolted to a corresponding flange on the end of the crankshaft by eight 14-mm. bolts.

Valves and valve gear.—The twin inlet and exhaust valves work vertically in the cylinder heads, and are operated as previously mentioned by rockers mounted on ball bearings, carried by supports screwed into the cylinder head.

The rocker levers operate the valve stems through hardened-steel rollers which are mounted on eccentric bolts. These, in conjunction with adjustable spherical joints screwed into the top ends of the vertical push rods on each side of the cylinders, give a fine adjustment for the tappet clearances. The spherical joints on the valve rockers are evidently provided to simplify the alignment of the valve-rocker supports which are screwed into the cylinder heads.

Semi-spherical joints are also provided at the bottom ends of the push rods, which work in steel cups inside the hollow tappets, and it should be noted that the hardened-steel rollers of the tappets are slightly offset from the camshaft centers, and each pair of tappet guides is held in position by a steel bridge clamp.

The dimensions of both the inlet and exhaust valves are the same, each valve weighing 1.25 pounds. The lift of the inlet valve equals 0.465 inch, and lift of the exhaust valve equals 0.443 inch. Clearance of inlet tappet equals 0.009 inch, exhaust 0.015 inch.

Camshafts.—The hollow camshafts each run in three plain phosphor-bronze bearings, and are arranged inside the top half of the crank chamber. The camshaft bearing bushes, which are hollow, are 53 mm. outside diameter, and are split on the camshaft axis and held together by wire rings on either side to allow the camshafts to be easily inserted in the crankcase. The bearings are located by grub screws screwed in from the outside of the crank chamber.

The camshafts are driven by gears from the intermediate gear wheel which meshes with the crankshaft distribution pinion, and arranged inside a casing formed at the rear end of the crankcase. The camshaft gears are

bolted to the camshafts by drilled flanges, which provide a vernier adjustment for setting the camshafts.

Half compression cams are arranged on the exhaust camshaft in the usual manner. The lateral movement of the floating camshaft is effected by a small lever at the rear end of the bookcase, which operates a two-start square-thread screw of 24 mm. pitch. The camshaft is returned to its normal running position by a spring arranged inside the front end of the hollow camshaft.

The half compression cams open the exhaust valves at 35 degrees E. and close at 23 degrees L.

Crankcase.—The method of cooling the interior of the crank chamber and sump indicates the fact that this matter has received most careful consideration in design. In the top half of the crankcase six of the seven main-bearing houses are cast so that the webs form air passages transversely across the engine. Two of these passages form the air intake passages for the two carburetors—similarly to those of the 160-H.P. Benz engines—as already stated.

The bottom half of the crank chamber is extended to form an oil sump and is of unusual design. The lower portion is cooled by 18 30-mm. aluminum tubes fixed transversely across the base chamber, the air being scooped into the cooling tubes by a large sheet-aluminum louvred cowl on the induction side of the engine. A corresponding but reversed cowl is fitted on the exhaust side.

Breathers are also fitted into the top half of the crank chamber. The simple type of wire spring clip fitted to the breathers should be noted.

Carburetion system.—Two separate carburetors are fitted, each feeding three cylinders. These are attached by flanges to the side of the top half of the crank chamber.

The pilot jet is formed by an extension of the brass tube into which the main jet is screwed.

This combined jet is screwed obliquely into the bottom of the float-chamber casting and not into the body of the carburetor. The float chamber is attached to the body of the carburetor by two bolts, and the throttle is of the horizontal-barrel type.

As already mentioned, the main air intake passages are cast in the crank chamber between the webs of the main bearings, air entering each carburetor through two ports cut in the end of the throttle, and also through the air passage at the base of the carburetor below the jets.

At full throttle most of the air is taken through the air ports in the throttle above the jets. When the throttle is closed for slow running the main jet is completely cut off, and all the air is taken through the passage containing the pilot jet. The slow running of the engine is very good.

The air intake ports in the end of the throttle are so designed that the petrol-air ratio remains practically constant over a wide range of throttle opening up to nearly half throttle; the last movement of the throttle, however, causes no increase in petrol flow, but, on the contrary, a slight decrease. No compensating arrangement is fitted for high altitude control. The body of the carburetor around the throttle is water-jacketed, and is connected by an arrangement of steel piping to the delivery of the water pump, and also to the lower water connections at the bottom of the water jackets between each pair of cylinders.

The design of the branched induction pipes is interesting. These are built up of aluminum tubes with cast aluminum bends, and weigh only 2.625 pounds each complete with connections. The internal diameter of the induction pipe equals 58 mm.

The attached R. A. F. test report computes the petrol consumption at 150 pints per hour, equaling 0.65 pint per brake horsepower hour.

The connections between the flanged bends of the induction pipes and

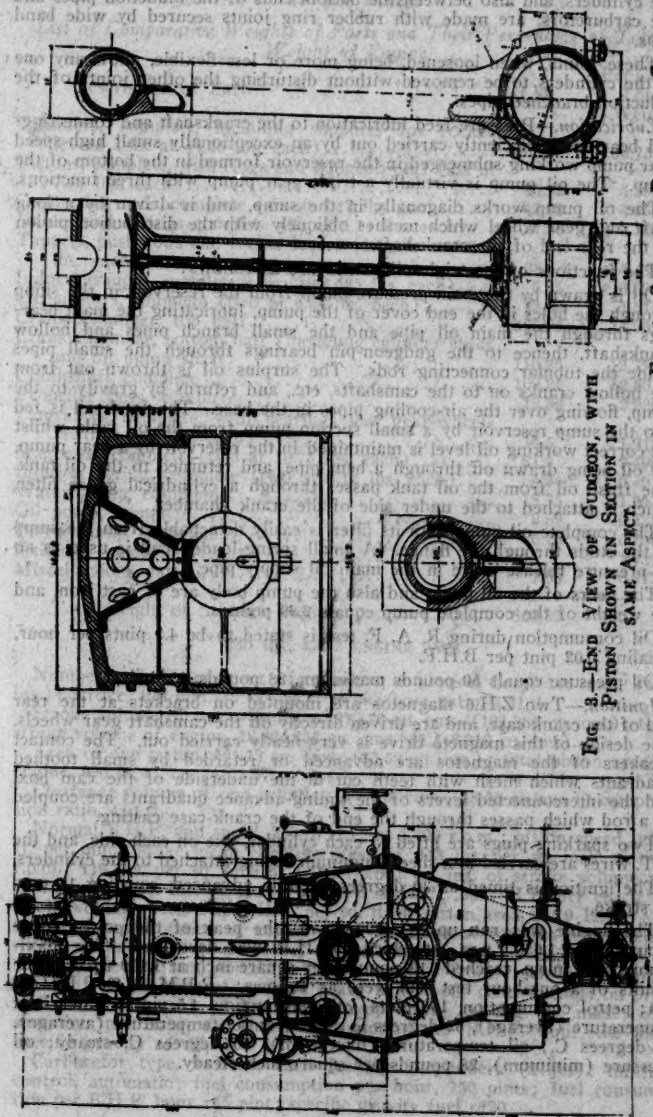

Fig. 2.—Cross Section Showing General Arrangement.

Fig. 3.—End View of Gudgeon, with Piston Shown in Section in same Aspect.

Fig. 4.—Sectional Details of Connecting Rod.

the cylinders, and also between the bottom ends of the induction pipes and the carburetors, are made with rubber ring joints secured by wide band clips.

These joints, when loosened, being more or less flexible, allow any one of the cylinders to be removed without disturbing the other joints of the induction branched pipes.

Lubrication.—Pressure-feed lubrication to the crankshaft and connecting-rod bearings is efficiently carried out by an exceptionally small high-speed gear pump working submerged in the reservoir formed in the bottom of the sump. The oil pump is virtually a triple gear pump with three functions.

The oil pump works diagonally in the sump, and is driven by a long shaft and gear wheel which meshes obliquely with the distribution pinion on the rear end of the crankshaft.

The functions of the triple oil pumps are as follows:

Oil is drawn by the main pressure pump from the reservoir in the sump through the holes in the end cover of the pump, lubricating the main bearings through the main oil pipe and the small branch pipes and hollow crankshaft, thence to the gudgeon-pin bearings through the small pipes inside the tubular connecting rods. The surplus oil is thrown out from the hollow cranks on to the camshafts, etc., and returns by gravity to the sump, flowing over the air-cooling pipes in the base. The fresh oil is fed into the sump reservoir by a small suction pump from the oil tank, whilst the correct working oil level is maintained in the reservoir by a gear pump, the oil being drawn off through a bent pipe, and returned to the oil tank. The fresh oil from the oil tank passes through a cylindrical gauze filter, which is attached to the under side of the crank chamber.

The complete oil pump with its filter is easily detachable from the sump by the studs through the flange. A small spring-loaded ball is used as an oil-pressure release valve in the main oil supply pipe.

The gears of the oil pump and also the pump body are of cast iron, and the weight of the complete pump equals 2.89 pounds.

Oil consumption during R. A. F. test is stated to be 4.5 pints per hour, equaling 0.02 pint per B.H.P.

Oil pressure equals 50 pounds maximum, 28 pounds normal.

Ignition.—Two Z.H.6 magnetos are mounted on brackets at the rear end of the crank case, and are driven directly off the camshaft gear wheels. The design of this magneto drive is very neatly carried out. The contact breakers of the magnetos are advanced or retarded by small toothed quadrants which mesh with teeth cut in the underside of the cam box, and the interconnected levers of the timing-advance quadrants are coupled by a rod which passes through the end of the crank-case casting.

Two sparking plugs are fitted to each cylinder, one on each side, and the H.T. wires are carried in a sheet-aluminum casing attached to the cylinders.

The ignition is timed as 30 degrees E. fully advanced, equaling 18 mm. on stroke.

The engine was run up to 1,700 r.p.m., the peak of the power curve occurring at 1,650 r.p.m., at which 250 H.P. was developed. The maximum brake m.e.p. reached 119 pounds per square inch at 1,100 r.p.m. The results of a one-hour test run were as follows: R.P.M., 1,400; B.H.P., 229; petrol consumption, 150 pints; oil consumption, 4.5 pints; water inlet temperature (average), 62 degrees C.; water outlet temperature (average), 71 degrees C.; oil temperature (maximum), 50 degrees C. steady; oil pressure (minimum), 28 pounds per square inch steady.

230 H.P. BENZ.

List of Comparative Weights of Parts and Their Percentage of Total Weight of Engine.

Description of Part.	Lbs.	Per Cent. Total Weight.
Cylinders, complete, including supports for rocker levers (six at 44.25 lbs. each)	265.5	30.753
Pistons, complete, with rings and gudgeon pins (six at 7.62 lbs. each)	45.72	5.296
Connecting rods, complete (six at 7.13 lbs. each)	42.78	4.955
Valves and springs complete (24 at 1.14 lbs. each)	27.36	3.169
Tappets, push rods and rocker levers (12 sets at 3.2 lbs. each)	38.4	4.447
Induction pipes, complete (two at 2.625 lbs. each)	5.25	0.608
Carburetors, complete (two at 6.54 lbs. each)	13.08	1.515
Crank Chamber, top half	103.25	11.959
Crank Chamber, bottom half (including cylinder studs and oil pump shaft)	110.25	12.77
Crankshaft complete	109.25	12.66
Propeller boss, complete	19.32	2.236
Camshafts, complete, with bearings less gear-wheels (two at 8.25 lbs. each)	16.5	1.912
Magnetos, complete (two at 10.75 lbs. each)	21.5	2.490
Ignition wiring, complete (two at 2.12 lbs. each)	4.24	0.491
Water pump, complete	8.75	1.013
Oil pump, complete	2.89	0.335
Petrol pump, complete	6.75	0.782
Exhaust manifold	15	1.737
Miscellaneous parts	7.53	0.872
Total weight of complete engine	863.32	100

230 H.P. BENZ ENGINE DATA.

Number and arrangement of cylinders, 6 vertical bore, 145 mm. (5.71 ins.); stroke, 190 mm. (7.48 ins.); stroke/bore ratio, 1.31:1.

Stroke volume of one cylinder, 3,137.476 cu. cms. (191.386 cu. ins.); total stroke volume of engine, 18,824.856 cu. cms. (1,148.316 cu. ins.).

Area of one piston, 165.13 sq. cms. (25.59 sq. ins.); total piston area of engine, 990.78 sq. cm. (153.57 sq. ins.).

Clearance volume of one cylinder, 796 cu. cms. (48.66 cu. ins.); compression ratio, 4.91:1.

Normal B.H.P. and speed, 230 B.H.P. at 1,400 r.p.m.; piston speed, 1,744 ft. per min.; brake mean effective pressure, 113 lbs. per sq. inch at 1,400 r.p.m.; 119 lbs. per sq. inch at 1,100 r.p.m.; cu. ins. of stroke volume per B.H.P., 4.99; sq. ins. of piston area per B.H.P., .667; H.P. per cu. ft. of stroke volume, 346.3 H.P.; H.P. per sq. ft. of piston area, 215.9 H.P.

Direction of rotation of crank, r.h.t.; direction of rotation of propeller, r.h.t.; normal speed of propeller, engine speed.

Lubrication system, forced to main bearings from reservoir in sump; brand of oil, 50 per cent. vacuum heavy and 50 per cent. sternol; oil pressure, 28 lbs. normal, 50 lbs. max.; temperature, 50 deg. C., max.; consumption per hour, 4.5 pints; consumption per B.H.P. hour, .02 pint; specific gravity of oil, .9.

Carburetor type, 2 Benz, each feeding 3 cylinders (2 jet); mixture control, automatic; fuel consumption per hour, 150 pints; fuel consumption per B.H.P. hour, .65 pint; specific gravity fuel, .720.

Magneto type, 2 Bosch Z.H.6; firing sequence of engine, prop: 1-5-3-6-2-4; numbering of cylinders, prop: 1-2-3-4-5-6; speed of magneto, two-thirds engine speed; direction of rotation of magneto, facing driving end of armature, anti-clockwise; magneto timing, 30 deg. 18 mm. on stroke.

Inlet valve opens, deg. on crank, 10 deg. E., inlet valve closes, deg. on crank, 55 deg. L.; maximum lift of inlet valve = h, 11.8 mm. (.465 in.); diameter of inlet valve = (smallest dia.), two of 52 mm.; area of inlet valve ports = $\pi \times d \times h$, 19.27 sq. cm. each; mean gas velocity through inlet valve, 124 ft. per sec.; clearance of inlet tappet, .009 in.

Exhaust valve opens, deg. on crank, 60 deg. E.; exhaust valve closes, deg. on crank, 20 deg. L.; maximum lift of exhaust valve = h, 11.25 mm. (.443 in.); diameter of exhaust valve ports = d (smallest diam.), two of 52 mm. dia.; area of exhaust valve ports = $\pi \times d \times h$, 18.37 sq. cms. (each) = 2.84 sq. ins.; clearance of exhaust tappet, .015 in.; direction of rotation of revolution counter drive, facing driving shaft on engine, anti-clockwise; speed of revolution counter drive, ½ E. S.

Weight engine complete with propeller hub, less water, fuel and oil and exhaust manifold, 848.32 lbs.; weight per B.H.P., ditto, 3.68 lbs.; weight of exhaust manifold, 15 lbs.; weight of oil carried in engine, 18 lbs.; jacket capacity of one cylinder, 1,873 c.c.; weight of water carried in engine, 30.9 lbs.; weight of radiators, less water, 136 lbs.; weight of fuel per hour, 135 lbs.; weight of oil per hour, 5.06 lbs.; total weight of fuel and oil per hour, 140.06 lbs.

Gross weight of engine in running order, less fuel and oil, cooling system at .65 lb. per B.H.P., 996 lbs.; weight per B.H.P., ditto, 433 lbs.

Gross weight of engine in running order with fuel and oil for six hours (tankage reckoned at 10 per cent. weight of fuel and oil), 1,920.89 lbs.; weight per B.H.P. with fuel and oil for six hours, 8.35 lbs.

THE TECHNICAL HISTORY OF THE AIRPLANE.*

By Captain F. M. Green, A.F.Ae.S., A.M.I.C.E.

The history and development of mechanical flight may be confined almost entirely to the present century. In the last century some success had been achieved in gliding flight, model airplanes had been made and flown, and at least two man-carrying airplanes had actually left the ground, but it was not until the year 1903 that the Wright Bros., having ended a painstaking research in gliding flight, earned for themselves the honor of being the first men to fly.

Directly the success of the Wright Bros. became known, or rather when people began to believe that what the Wright Bros. claimed was actually true, the development of flying became rapid in the extreme. Today one is tempted to wonder why it was that it had taken man so many centuries of civilization to do what now seems to most of us rather an ordinary thing. We can call to mind nothing in the nature of an invention, nothing new in first principles. We hear of flights from Turin to London, of airplanes that carry loads that are measured in tons, we know that flying is common at heights of four miles or more, where the air is less than half its usual density. When we hear of these things it seems as unbelievable to us now that until fifteen years ago no one had flown for half a mile carrying just his own weight as it seemed unbelievable to us fifteen years ago that we should ever fly at all.

The chief reason why the development has been so rapid is undoubtedly that, until an airplane had been made to fly at all, experimenting on full

*Paper read before the Aeronautical Society of Great Britain.

scale was impossible. It reminds us of the advice of the old lady to her grandson, "Don't go near the water until you have learned to swim." Directly mechanical flight was possible, full-scale experiments were carried out, often at the risk of the life of the experimenter, and the more obvious problems became apparent. All the skill of the engineer and all the resources of science were called in to solve the problems as they occurred, until in the short space of fourteen years our scientific knowledge of aeronautics compares very favorably with the world-old science of navigation of the sea.

It is the intention of this paper to discuss the progress that has been made in dynamic flight, and to consider the chief factors that have made this progress possible. An attempt will be made to state in a general way the standard of development as it exists today, and to consider the possible direction of progress in the future.

TYPES OF MACHINES.

The flying machine is an apparatus in which an arrangement of surfaces is moved in such a way that the reaction of the air provides a vertical component of the force on the surfaces equal to the weight of the apparatus. An airplane, as we know it, has its supporting surfaces fixed relative to itself and is driven forward by the reaction of one or more airscrews.

Besides an airplane, two other sorts of flying machines have offered possibilities to the inventor. One is known as the Helicopter, in which the lifting surfaces are revolved in a horizontal plane. The second is called an Ornithopter, and in this the supporting surfaces are moved in a way to imitate a bird flapping its wings. Although both these types are conceivably possible, their usefulness and practicability appear doubtful, and it is only proposed in this paper to consider the airplane as we know it.

AIRPLANE PERFORMANCE.

At the beginning of flying, the mere fact that an airplane left the ground at all was considered sufficient proof of its excellence. Nowadays we make careful measurements of the characteristics of each new airplane at various heights, and the word "performance" is used to refer to the speed, rate of climb and load carried. A good performance is one of the important essentials of usefulness of an airplane, and we will now consider the factors that have enabled us to improve so much on the early airplanes.

There are four main factors that govern the performance of an airplane. They are:

1. The wing surface that supports the weight;
2. The total weight supported in flight;
3. The resistance of the airplane to forward motion;
4. The thrust available from the airscrew or airscrews.

These factors are very intimately connected, and each in turn will be considered, its effects discussed, and the possibility of improvement outlined.

SUPPORTING SURFACE.

Considering the first factor, the wings, it is necessary to know how to express the value of any particular arrangement of surfaces. The result at which to aim is to obtain the maximum vertical reaction or lift for the minimum horizontal reaction or drag. The term lift drag ratio is, therefore, a measure of the effectiveness of a plane. Another attribute is, however, desirable, viz: that the lift per square foot of plane shall be high, so that as little surface as possible need be used for a given minimum speed of flight.

Planes vary in curve, plan form and arrangement and number of surfaces. The designer will vary these factors according to the sort of airplane he wishes to make.

WING SECTION.

The Wright Bros. did not know if they would be able to fly at all. They chose deeply cambered wing sections so that the maximum lift could be obtained at the minimum speed. The lift drag ratio of their planes was probably about 12 to 1, very much better than still earlier experimenters could obtain with flat planes. Present-day wing sections enable us to get a lift drag ratio of 17 to 18, while model tests have shown as good a ratio as 23.

This has been done at some sacrifice of maximum lift coefficient; that is to say, for a given minimum speed we need more surface to support a given weight than with a more cambered plane with a worse lift drag ratio. It would be an advantage if we could vary the camber of a plane to suit the speed, and many atempts to do this on an airplane in flight have been made. Unfortunately, apart from increased complication, the increased weight is likely to cause greater loss than gain.

Shortly before the war an experimental airplane was made at the Royal Aircraft Factory, which was fitted with wing flaps nearly the whole length of the plane. These flaps could either be turned down for low speed, or up for high speed, or they could be used differentially for balancing. It was, in fact, rather a crude way of varying camber. Although it achieved the distinction of being possibly the fastest airplane of the day, it was eventually considered that the extra weight and complication did not justify the device.

From many considerations it appears that much improvement in lift drag ratio is unlikely, but when the speed of airplanes becomes very much greater than it is today, it is quite possible that some device for changing the section of a plane during flight will be found necessary to obtain a reasonably slow landing speed.

PLAN FORM.

Planes are generally made nearly rectangular in plan, but with the ends more or less rounded. The ratio of span to chord is called aspect ratio, and the higher this is made the better will be the lift drag ratio. Here again it is necessary to compromise, as the bigger the span of the airplane the heavier will be the wings, the greater the resistance of the necessary bracing and the rapidity of the maneuvering of the whole airplane will be decreased. From practical consideration, therefore, the aspect ratio of an airplane is generally from 5 to 8; recently some large airplanes have gone considerably beyond this figure, but it is very doubtful if anything has been gained.

Little is known as to the effect of using planes other than roughly rectangular, but it is not very likely that much improvement will be found with planes of other shapes.

ARRANGEMENT OF SURFACES.

The original airplane of the Wright Bros. was a biplane, consisting of two planes separated by about the width of the chord. Shortly after, monoplanes, triplanes and quadruplanes were made and flown. It is fairly certain that the most efficient plane is the monoplane, for there is no possibility of interference of the planes as there is on the multiplane. A biplane, however, is probably only a little worse than the monoplane, while a triplane is a little worse than a biplane, and so on. Structurally, the biplane is very much easier to make than the monoplane, and, owing to this, the biplane construction has been almost universally adopted.

Triplanes and quadruplanes have not had very much success, although it is quite possible that for very large airplanes the advantages gained in construction may outweigh the disadvantages of their somewhat low effectiveness.

From time to time suggestions have also been made that the amount of wing surfaces should be varied. This again is a complicated thing to do, and with modern wing sections there is practically never any gain in reducing wing surfaces; in fact, most airplanes would fly faster and not slower by having more surface if such surface could be introduced without extra weight or external resistance. This applies particularly to flying at heights.

WEIGHT.

The total weight that is carried on an airplane is obviously all-important, and it will, therefore, be useful to see what prospects we have of decreasing weight of airplanes of any given type. We will divide the airplane into the following components:

Useful load;
Motor unit;
Fuel, oil and tanks;
Remainder, being airplane parts proper.

The object of the airplane flying at all is, of course, to carry a useful load for a certain distance. In this load we do not include fuel and oil, and as the tanks vary in weight with the amount that has to be carried, it is convenient to include in the fuel and oil the weight of the necessary tanks and pipes.

For the purpose of general analysis it is assumed that the motors consume the same amount of petrol per B.H.P. per hour, and the figure 0.7 pound per H.P. hour, is sufficiently accurate to cover the weight of fuel, oil, tanks and pipes. There are, of course, engines which do better than this, and many which do considerably worse. It is, however, a fairly good average for motors as they are now.

Analyzing a number of airplanes that have been used under active service conditions it has been found that, apart from the useful load, the motor unit, and the petrol, oil and tanks—the remainder representing the airplane proper—is very nearly one-third of the whole weight—32 per cent. being rather nearer. This weight includes the wing structure complete, the body complete with its fairing, floor, seats, controls, instruments, etc., the tail plane, elevator, rudder and fin, and the landing gear and tail skid. This weight is on the average made up as follows:

	Percentage of airplane parts. Per cent.	Of total weight. Per cent.
Wing structure complete	41	13
Body complete	41	13
Tail plane, elevator, rudder and fin	6	2
Landing gear	12	4

If we take 32 per cent. as the weight of the airplane proper we have the balance of 68 per cent. to divide up among the motor unit and its fuel and the useful load. Various diagrams were then shown depicting the combined weight per H.P. of the motor unit with its fuel, oil and tanks for various times of flight.

We have now only to decide what percentage of useful load we shall carry, and for what time it is to be carried, and we can find at once the weight per H.P. of the complete machine.

POSSIBILITIES OF REDUCTION OF WEIGHT.

The weights of the airplane parts have now been fined down to a considerable extent, and, except for the use of newer materials, progress in

saving weight is not likely to be very pronounced. Undoubtedly as time goes on we shall learn to make equally strong structures for less weight, and the development of aluminum alloys and pressed-metal construction may help to a marked extent. At the present moment wood is used very largely in the construction of both the bodies and wings. To a great extent this is due to war urgency, and no doubt airplanes of the future will be made of more durable materials.

The chief particular in which it is probable that weight will be reduced is in the weight per H.P. of the motor. Although progress in this respect has been remarkable, it is probable that we have not nearly reached the limit of weight in internal-combustion motors. At the present day the complete motor unit—*i. e.*, the motor itself, the airscrew, water and radiator if water-cooled, exhaust pipes, and so on—weighs generally from 3 to 4 pounds per H.P. Certain very light motors have been made in which the weight is very little more than 2 pounds per H.P. It is quite conceivable that in the next few years motor units of 1½ pounds per H.P. will be available, and in all possibility with the improvement in material will enable us to make motor units weighing not more than 1 pound per H.P. in the more distant future.

With regard to fuel consumption, the best petrol motors have a brake thermal of efficiency of about 30 per cent. It may be this will be improved, and here again saving of weight may be effected.

The only motive power used today is supplied by internal-combustion engines. Some day no doubt a lighter form of motive power will be developed, and it is in this respect that the most startling progress may be expected.

RESISTANCE OF THE AIRPLANE TO FORWARD MOVEMENT.

The resistance of the airplane is made up of two factors. The first is the resistance of the main planes. The second item is made up of the resistance of the body, the struts, wires and fittings of the wings, the landing gear, tail plane, rudder, and so forth. These can all be grouped together under the general term of body resistance. It is in respect to the reduction of this resistance more than anything else that the modern-day airplane owes its superiority over that of the original Wright machine.

The first man who seems to have understood the importance of reduction of resistance was the late Edouard Nieuport, who in 1909 produced an airplane with a very fat body which almost completely enclosed the pilot and succeeded in making a big improvement in speed on an airplane of comparatively small power. Since that day all modern airplanes have bodies which are enclosed as completely as the requirements of the pilot's view allows.

DETAILS OF RESISTANCE.

The resistance of the external parts of the airplane has now been fined down to a considerable extent; modern-day airplanes do not differ very much in this respect. The various components of the airplane will generally be found to have the following proportional resistances:

	Per cent.
Body	62
Landing gear	16
Tail plane, fin, rudder and elevator	7
Wires, struts and fittings of the wing structure	15

It will be noted that by far the larger part of this resistance is made up of the body itself. A large part of the body resistance, however, is really used in providing cooling for the engine, whether it is an air-cooled or water-cooled engine. Whether the radiator is let into the body or is external to it does not appear to make a great deal of difference. The body

is also of greater resistance than it need be, as it is necessary that the pilot shall have a good view and shall not be entirely enclosed. Bodies round in section are somewhat lower in resistance than the square bodies, but the difficulties of manufacture generally make their use undesirable. It is not likely that we shall be able to reduce the body resistance to a very great extent unless we can use a motor which does not require external cooling. It is true that it would be possible to use the complete surface of the body or of the wings as a radiator, and by this means the waste of power would be negligible, but in order to do this the radiator becomes an impracticable and heavy piece of apparatus.

The resistance of the landing gear is considerable, and it is difficult to see how this can be very much reduced. It has been suggested by many people that the landing gear could be stowed away inside the body while the airplane is in flight, and no doubt in the future this may be done. It is, however, very difficult to design a folding landing gear without an undue sacrifice of weight or a large increase in the cross section of the body.

Another factor which influences the resistance of the airplane to a considerable extent is the increased resistance of the parts of the airplane that are in the slip stream of the airscrew. The allowance to be made for this is not known very definitely, as in either a tractor or pusher airplane the resistance of the body itself tends to increase the efficiency of the airscrew.

THRUST AVAILABLE FOR FLIGHT.

The thrust available from the airscrew varies directly as the horsepower of the motor, directly as the airscrew efficiency, and inversely, as the forward speed of the airplane. If the efficiency is constant the thrust available at any speed will depend upon the horsepower of the motor, consequently the weight per H.P. of the whole airplane is an expression for the ratio of thrust to weight of airplane. I have referred to a general method for obtaining the proportional weight per H.P. of an airplane knowing the weight per H.P. of the motor, the number of hours of flight, and the percentage of useful load that has to be carried. As in practice the efficiency of the airscrew does not vary much, and as the resistance of modern-day airplanes per pound of weight is fairly similar, it is possible to predict rather roughly the performance of the airplane simply from the weight per H.P. figure.

AIRSCREW EFFICIENCY.

We have said that airscrew efficiency does not vary much in practice. There is, however, a considerable difference between the best attainable efficiency and that obtained under certain conditions.

As Lanchester has shown in Aerodynamics, for maximum airscrew efficiency the pitch must be a little greater than the diameter. From first principles also it is easy to see that the diameter must be a certain minimum so that the loss in the slip stream shall be small. Now it is always convenient to have an airscrew as small as possible, for when it is smaller it will be lighter and less in the way.

Again, unlike the screw of a ship, the airscrew has to work at bigger thrusts when the airplane is going slow and climbing than when it is going at full speed on the level. As a consequence, the airscrew of a modern airplane nearly always has its maximum efficiency when its effective pitch is at its largest—*i. e.*, when it is going fast, and its efficiency is lowest when the pitch is small, that is, when the airplane is climbing.

Another disadvantage when the airplane is climbing is that the engine is more heavily loaded, and in consequence the airscrew goes slower. With engines as they are this always means less horsepower. We need, therefore, either a change speed gear or an airscrew in which the angle of the

blades can be changed. This will avoid a drop in horsepower, and to a lesser extent the loss of efficiency when climbing. We also need, however, a propeller of varying diameter, so that combined with a change of angle and a suitable engine speed we should get nearly a maximum efficiency under all conditions.

There is not much room for improvement in airscrew efficiency working under the best conditions, as the efficiency is as high as 85 per cent. When climbing, however, the efficiency will frequently drop to less than 65 per cent.

Attempts at variable-pitch or variable-angle airscrews have been made, but none seem to have reached a practicable stage. The design of such a screw is quite possible, and no particular invention is required to make it a success. As the demand undoubtedly exists, variable-pitch airscrews will be made successfully, and will become a standard part of the airplane equipment in the near future in all probability.

PERFORMANCES OF THE FUTURE.

The airplane flown by the Wright Bros. had a speed of 30 to 35 miles per hour, and we have in a few years made airplanes that fly four times as fast. What, then, are the prospects of the future?

Before considering this we must consider the effect of height on the speed at which an airplane flies.

Very briefly, the resistance of the airplane and the horsepower of the engine each fall off when the density of the air decreases, but with engines as now designed the horsepower falls off faster than the resistance.

It is now proposed to consider what speed we are likely to attain in an airplane of the future flying at a height of 10,000 feet. As it is a hypothetical airplane engine, we are entitled to assume that all the improvements outlined in the preceding paragraph are incorporated into it. The assumptions we will make are:

(*a*) That the planes have variable camber.

(*b*) That the motor unit weighs 2 pounds per H.P., and will maintain this horsepower at all heights up to 10,000 feet.

(*c*) That the body is the best possible shape, completely enclosed, with no external fittings.

(*d*) That the landing gear shall be folded inside the body during flight.

(*e*) That the airscrew efficiency shall be 85 per cent.

(*f*) That the percentage of useful load carried shall be 15 per cent.

(*g*) That the duration of the flight shall be two hours.

Assuming that the weight of the structure is as at present, the weight per H.P. of the complete airplane will be 6.4 pounds. As we are considering a faster airplane than is at present made, it may be as well to allow a little more weight for the airplane structure, and we will assume that the weight per H.P. is 7 pounds.

We will now turn the percentage weights into pounds by assuming that the useful load is to be 450 pounds, which may be two people and luggage or perhaps one man and mails. The gross weight of the airplane will therefore be 3,000 pounds and the H.P. of the motor 430. We will imagine that the airplane is of a type in which there is practically no resistance due to the slip stream of the airscrew; also that the motor requires no external cooling. In order to accommodate the engine and crew we will consider that the biggest cross section of the body is 12 square feet, approximately round in section. The resistance of such a body will be about 2 pounds to the square foot at 100 feet a second forward—*i. e.*, 24 pounds in all.

The resistance of the rest of the airplane by careful design can be kept down to 16 pounds at 100 feet a second, consequently at that speed the total resistance other than the wings will be 40 pounds.

The area of the wings necessary will be governed by the speed at which it is considered reasonable to land; taking 60 miles per hour as a possible speed and assuming that the planes can be converted into a deeply cambered section for landing, the permissible loading of the airplane will be 15 pounds per square foot, making a total surface of 200 square feet.

We will now take a speed of 350 feet a second, or 240 miles per hour, and find the horsepower required at a height of 10,000 feet—*i. e.*, where the density of the air is .74 of the normal. The wing drag may be taken as a fifteenth of the weight, as with wings of varying camber we shall be able to get a high lift drag ratio even at a small lift coefficient. The resistance due to wing drag, therefore, will be 3,000 ÷ 15, which is 200 pounds.

As the airplane is flying at 350 feet a second, the body drag will be 3.52 times what it was at 100 feet a second, which will be 490 pounds. As, however, the air is 26 per cent less dense, its resistance will be reduced to 364 pounds.

The total drag will be the sum of the body drag and the wing drag, 564 pounds in all. Taking an airscrew efficiency of 85 per cent. on the power required for flying, it will be very nearly 430, which is just the power available.

The example just taken may seem fantastic to many people here tonight; it must be remembered, however, that when the Wright Bros. were flying at 40 miles per hour, speeds of 150 may have seemed to them equally fantastic. On the assumptions we have made it is reasonably certain that a speed of four miles per minute can be obtained, and as motors of less weight per H.P. become available there is no reason why this speed should not be exceeded.

The rate of progress in increase in speed will depend upon the demand for speed. If war were to last it is almost certain that speeds of much more than 200 miles per hour would be common in quite a few years from now.

STABILITY AND CONTROL.

In the original Wright airplanes no attempt was made to obtain stability. The airplane was definitely unstable and required considerable care on the part of the pilot to keep it in equilibrium.

The early French airplanes were a little better in this respect, but the first airplane to prove that it really was stable—*i. e.*, it could fly without controls—was that designed by Lieut. Dunne. This was an airplane of somewhat curious construction, and never became a success, although it certainly was stable; on a windy day it was somewhat difficult to persuade it to fly on a straight course, and its control when landing seems to have been insufficient. Lieut. Dunne, however, certainly showed that a completely stable airplane was a possibility.

About the time at which Dunne was making his experiments certain German designers claimed that they had produced a stable airplane. No record seems to exist of any test that proved their airplanes were stable, and in any case they certainly suffered from lack of control. The first airplane of normal type that was proved to be stable was produced as the result of experiments carried out by the late Edward Busk. An airplane known as the R. E. 1 was used for the first experiments, and it was modified until it was made completely stable. Tests were carried out in order to prove that this airplane really was stable, and with Busk as pilot, carrying Colonel Sykes as observer, a flight was made from Farnborough to Salisbury Plain, in which the only control used was the rudder, and this, for part of the time at least, was worked by Colonel Sykes.

Since that day most English airplanes have been designed to be stable; it has been found that for purposes of fighting an airplane that is extremely controllable is necessary, and it has been also found that if an

airplane is made too stable the controls are apt to become too heavy to make the airplane sufficiently sensitive. In modern-day airplanes intended for war purposes the aim of most designers is to make the machine just comfortably stable over the range of speeds used in ordinary flight, but to provide sufficiently large controls that the inherent stability of the airplane can be entirely overcome.

It is often asked if we have stable airplanes why do we still have flying accidents? It would be quite possible to design an airplane in such a way that whatever the pilot did with the controls he would be unable to upset the equilibrium of the airplane. Such an airplane, however, would be valueless for fighting, and as practically all airplanes nowadays are made for war purposes there must continue to be an element of danger for the unskilled pilot.

STRUCTURAL STRENGTH AND SIZE.

Increase in speed and the increase in controllability both tend to increase the stress that will be put on the structure of an airplane in flight. Experiments have been carried out to determine the increased loads that are likely to occur when doing sharply banked turns and loops. It has been found that the load is frequently increased to three or four times the normal.

Present-day airplanes have to be very much stronger than before. Modern-day machines that are expected to maneuver rapidly are not considered tolerably safe unless the wings can support at least six times the weight of the airplane. Fortunately, large airplanes for peaceful purposes will not be required to maneuver very rapidly. This fact will have an important bearing on the possibilities of aeronautics.

In an earlier paragraph it was stated that the proportional weight of the airplane parts proper was more or less constant. This applied only to machines of the sizes that are used now—*i. e.*, up to about six tons loaded weight. As airplanes are made bigger, if they are designed on similar lines, the weight of the wing structure per square foot will increase, assuming they are to be designed for the same factor of safety. It is true that on the large airplanes now made this increase of weight is not very apparent because greater trouble is taken to economize in the design of the wings. This increase of weight, however, is likely to be important for machines of much larger size, but as we shall be able to use somewhat lower factors of safety the weight of the wing structure will not increase as rapidly as it would otherwise have done. There certainly is a limit of size to which it is possible to build an airplane; what this size will be the future will show. It is safe, however, to say there will be no serious difficulties in building airplanes two or three times as big as the largest we have considered.

CONCLUSION.

The first few years in the history of flying have been years of remarkable progress, and we are perhaps likely to be self-satisfied. We have no logical right to think that our flying machines of today are more than the crudest caricatures of the airplanes of the future. We engineers are a little apt to become so engrossed in our immediate difficulties of design that we do not see further ahead than is suggested by our own ideas, and make predictions for the future that only follow along the lines of progress that we can clearly see.

The writer has often been asked if he thinks that flying will ever become an ordinary means of travel. His answer is that he does not think but that he knows it will become so. Ten years ago to attempt to fly from place to place was considered suicide; five years ago it was an adventure; today to many of us it is a very ordinary affair. When we get an opportunity to use the airplane in preference to the motor car, the train, and the ship, we

travel at a miserable 100 miles per hour or so in a noisy, cold and drafty construction made mostly of canvas, wood and glue; yet we reach our journey's end far quicker and in a happier state of mind than by any other means. A few years more and we shall laugh to think of the airplanes of which we are now so proud, we shall shudder at the risks we ran, and we shall travel with comfort, speed and safety.—"Aviation."

AERIAL TRANSPORT AND TRAVEL.*

By Lieutenant Colonel Mervyn O'Gorman, C.B., D.Sc.

There is talk today of aircraft for transport and travel. The carriage of goods, mails and people, the exploration of remote districts, the conduct of photographic surveys, the searching out of valuable trees in pathless forests, the speedy conveyance of officials to their administrations in distant climes—these things and others are everywhere hopefully dwelt on by the imaginative among those engaged in air work. It would be grossly unjust to say that such hopes and fancies detract from the energies which these persons expend on their war duties; they rather indicate the intention of winning and of carrying on thereafter the life of a virile community.

In a paper before the Aeronautical Society, in June this year (see Nov. 1, 1917, issue of "Aviation and Aeronautical Engineering"), I gave reasons to show that in the interest of the war we must care about the future of transport by aircraft. I will recapitulate the position under *A*, *B* and *C*, for there are three stages in the logic of the matter.

THE THREE STAGES.

A. We are to have an Air Ministry and an Air Force: the Act shows that they are not temporary—a token from which alone we may say that we intend to have a fighting Air Fleet.

B. An air fleet differs from a sea fleet in being much more easily expendible. No one would dream of rebuilding the British High Seas Fleet six or eight times per year of war, but an air fleet in action requires, and will continue to require, at least this. Thus the airplane has in this context the interesting singularity of being intermediate between a cartridge and a battleship. It is not quite so rapidly expendible as the former, but it is an expendible munition, and when we decide to have an air fleet we also decide, in logic, on the third conclusion, *C*.

C. This corollary is that we will maintain an aircraft construction organization in peace that shall be competent to produce some six or eight air fleets per annum in war.

At present we have such an industry—not large enough, perhaps, but we have at least a live and flourishing plant which is bearing fruit, which can grow, and which, be it noted, can also expire. A living organism can be killed by deprival of oxygen. In three minutes a man is drowned, and 40 years' training of brain and hand can be thus quickly reduced to an inert mass. Similarly rapid is the rate of disintegration of a highly organized technical production department like the Aircraft Industry if it be starved. Today war orders continue at full flood to fill up the wastage. If they stop suddenly we shall find within three months that the shop organizations are broken up, that all that remains to us is the husk; we shall realize that we could, by timely measures, have saved much of our expensively purchased experience and organization, and have retained a value far greater than the mere buildings and plant. Once the designers and

* A lecture delivered before the Royal Society of Arts.

workmen are scattered, that which is so difficult and expensive to build up will be gone. Our capacity for the production of aircraft will be an object of derision, like a bouquet of hairs in an old broom.

This simile will have succeeded if it has led a few score of persons to inquire, "What on earth are we to do to keep this mechanism of production in being without keeping up in peace the war rate of expenditure?" A part of the answer was outlined in my first sentence; we must so utilize aircraft on commercial duty, and so nurse it in its days of trouble, that it shall itself earn the best part of its keep.

If aircraft using can be induced to pay, aircraft making will of itself continue. But, in spite of the firmest belief in the value of aerial travel, the mere institution of services takes time. Neither three months, nor six months, nor a year, will see a thriving mercantile air fleet engaged on its routine duties and issuing its routine orders on which the construction business depends. Yet a less period of inaction than this will extinguish the industry. This is the period I ventured to call the "hiatus" in the discussion on Mr. Holt Thomas's historic lecture last May. To bridge the hiatus we must do something *now*. There is no other alternative. No one expects a continued unrequited expenditure of from 25 to 50 million pounds per annum on aerial war material in peace time. On the other hand, no one would object to our making even this large amount of aircraft if it nearly earned its keep by remunerative services in any of the travel and transport businesses, or even if we could be assured that, after a period of Government support, it would eventually draw near to the standing of a self-supporting industry.

HISTORIC PRECEDENTS.

If we look back to the birth of the railway and the steamboat, we see that for some years they struggled against public apathy—though their potentiality for good and for the creation of wealth and trade was almost as great in Watt's time as now. This apathy, amounting to boycott, rendered them unremunerative for a long period, and deprived the world of travel, and joy and change, and fresh air, and trade progress for well nigh a generation. In the case of the road vehicle, which we now call a motor-car, we had something worse than apathy—we had legislation for the alleged protection of the public, all meant in kindness—which killed the motor-car in 1837 and retarded the car of 1892 till France was well ahead of us. In the case of dynamo electricity we underwent a process of protective legislation against shock and fire possibilities, which threw us well behind Germany and the United States. The story of legal impediments to scientific advance makes one wonder whether we are a free people—that is, with freedom to advance, or are we only set on freedom to stagnate?

The greatest danger that aircraft has to fear, after public apathy, is legislative interference. It is not that the British law, excepting only when it is panicky, is worse made or more malignantly administered than another. The kindly intentions of legislation towards the public are generally as laudable as our laws in technical matters are detrimental. Yet such a community as ours can only live and thrive on its technicians. We are prone to drown the puppy lest he should bite, instead of adopting our own unique maxim of allowing the law to lag, so that the puppy shall have one fair bite before he is muzzled.

ANTI-PROGRESSIVE LEGISLATION.

We now know that it was shameful to have waited for the war before building up our Department of Scientific and Industrial Research. No branch of administrative expenditure was known to pre-war politics whose function was to venture forth, or to encourage us by Press publicity, or

otherwise to advise the public to risk a little inconvenience during the period of evolution of a difficult art or science. The four-mile, the twelve-mile, and even the twenty-mile per hour motor-car speed limit prove that technical legislation has hampered and did not encourage. Of legislation relating to aerial transport it may safely be said that it is a case requiring the greatest delicacy of handling. It must be taken as certain that Government assistance, I do *not* say subsidies, for the industry must be forthcoming unless it is doomed, but even if this assistance were certain there remains cause for anxious thought today.

The danger of Government support lies in the conditions which a self-protective public is liable to impose before the technical possibilities are sufficiently known to form the basis for any sort of legislative interference. These conditions are liable to be all the more onerous since they could be imposed as of right, and regarded as a *quid pro quo* for the "*assistance.*" The fearful might say that great unknown risk to those on the ground (*omne ignotum pro magnifico*) is involved in any program of commercial aeronautics. The braggart has already talked of a sky darkened by airplanes, of the cerulean clotted with colliding aircraft, and the timid voter silently wonders how *he* is to be protected from the debris. Luckily we have the facts with which to dispel this apprehension. Today we may safely say that £50,000,000 worth of aircraft is yearly poured out from this country's factories. The large majority of this is flown to its destination, or used for tuition and defense at home, yet we still see the sun.

We have a full measure of the nuisance exaggerated by the more hurried training of fliers, and the extreme lightening of structure imposed by war conditions. Such nuisance as there is is purely local, and is really negligible. Now there is no prospect for many a long day of any such £50,000,000 output on commercial aerial transport. It is merely necessary to recall that our total expenditure on merchant shipping is a mere fraction of this to see the absurdity of apprehending and legislating as if we were to have this immense user of aircraft. There is no slightest probability of any nuisance in excess of what little we suffer today, even if we had, which I see we shall not have, an aerial fleet of this magnitude.

I have indicated the need for restraint in law making, but there is also need for positive official action, if the bells that ring in peace are not to sound the knell of aerial activity. The Allies must, during the war, frame a joint policy as to the tolerance of each other's trade and postal aircraft, and agree to air routes throughout the parts of the world which they control. With the large expanse of our colonies and our own industrial and postal importance, we have much to offer to the Allies which will be a valued equivalent for that which we shall obtain from them in the way of alighting rights. Being an island, and accepting, as we must accept, the proposition that the most significant section of aerial transport will be outside the confines of Britain, we need that our Allied neighbors shall extend, not only tolerance, but welcome to our machines when engaged on their transport work. We and our colonies should have to be prepared, and, I believe, are prepared, to extend a corresponding welcome to their airmen at home.

PREPARING FOR PEACE.

International agreements, even though simple in subject matter and non-controversial, are always long drawn out. Everyone who understands factory organization will agree that we cannot afford to wait until after the war for these parleyings to begin. Our aeronautical organism will assuredly become disintegrated too quickly. If we can get four or five willing men round a table to agree to admit aircraft mutually, to register them, to agree to simple preliminary rules of the road, and to agree that their respective countries should initiate a few lines of landing grounds so as to constitute safe routes, the early days of trade flying have little more to ask for from the foreign offices of the world.

No doubt in some four or five years of active aerial transport organization work the paltry fifty or hundred airplanes of our early post-war efforts at transport and travel will have grown up into a significant carrying trade for high-speed work, large quantities of aircraft will have been used for commercial purposes, and it will be found useful to introduce, and there will be experience enabling us effectively to introduce, valuable regulations for the protection of passengers in aircraft and the public on the ground beneath. At the present day it is clear to all who have eyes to see that no adequate knowledge exists on which to frame rules with discrimination and without grievously hampering technical development.

THE DANGER OF SAFETY.

I can imagine a legislature winning a round of applause from the *inane vulgus* by driving a hard bargain with the wretched trader who, in the face of starvation, might accept any regulations, however hampering, in exchange for the means to live. I can imagine the imposition of tests of airworthiness, strength tests, the enforcement of hobbies such as multiple engines, silencers, special alighting gears, the carrying of parachutes, fire extinguishers, wireless equipment for calling for aid, and countless other things, which may, no doubt, come in their own time, but which would clog the versatility of the engineer, and for years condemn to unproductiveness the nascent industry of air travel. The payment of an army of inspectors would be put down as encouragement money, and would be a charge against the aeronautical vote. A landowner would ask for the power to capture the flier who has most reluctantly alighted on his ground. He might ask for a law allowing him to impound the airplane, on the chance of his registered number being inauthentic, and so secure payment of the damages which he wishes to claim.

It is just possible that regulations of this sort might meet with support, as much as support is at present accorded to the 20-mile speed limit of the motor-car. They would be next to impossible to remove, though they might be demonstrably hampering the aircraft development of the country to an enormous extent. The intentions would be good, the desire would be to secure that only the best aircraft were used, and that the public suffered no injury by its use. It was for half a century thought that the public could suffer no injury by the four-mile speed limit being imposed on mechanical traction! Later it was thought that as a 12-mile an hour speed limit was ample for a coach-and-four, it must be an ample speed for a car, and so the automobile industry passed to France and Germany, in spite of the fact that the first car plied for hire in England early in the reign of Queen Victoria, when there was *no* speed limit.

A SAFE PROPHECY.

Let us rest assured that the amount of civil aircraft will for some years be so much less than the amount of military aircraft now existing, that the total of damage to property and inconvenience to the public, which we at present know to be small, will be much further reduced in proportion to the lesser numbers, the future technical advances, and the less strenuous conditions of maneuvering imposed on peace airplanes.

LABOR.

Labor.—Among the expectations based on aerial transport and travel are those of employment for individuals throughout the whole social scale as well as officers and men from the Army and Navy. Our producers, and therefore their employment of labor, depend on the bodies which employ aircraft—viz: (a) Primarily the Air Force; (b) foreign buyers, nations

and colonies; and (c) companies instituted for transport and travel, including postal duties. All these bodies need labor, and all are interested in continuing harmonious relations between those who direct the work and those who manually carry it out.

It has been suggested that since the aircraft industry is a new industry, an exceptional opportunity exists of introducing a totally new atmosphere of harmony unembittered by the old trade quarrels. This is most desirable, but the novelty of the technics of aircraft has not *in fact* eliminated the old trades at all; on the contrary, it has developed the demand for these "tradesmen." The Aircraft Industry is not a new trade in the sense in which that word is used by the world of labor in the phrase "Trades Union." The Aircraft Industry includes members of almost all the engineering trades, as well as important additional trades, such as woodworkers, carpenters, organ builders, cabinet makers, fabric workers, etc., and it is necessary for that harmonious relationship to be established between employer and employee, in spite of the fact that the history of all the old trade differences will unavoidably be also the history of the persons and corporate bodies imported into the new industry, with this addition, that a trade quarrel on the subject of organ pipes or on the piece-work price of chairs, may mean a stoppage of airplane work, owing to the newly imported craftsmen holding on to the old unions.

Again, it would be unintelligent to pretend that aircraft making can be developed under a compact between employees and managers which shall guarantee continuity of employment for all the manual workers in exchange for a continuity of labor supply to the other. The difficulty is a fundamental one, and perhaps is a worse difficulty in the Aircraft Industry than in most others. It is this. The technical developments of the art, the unforeseeable shortage of material, and the unavoidable changes of method and of materials, tend, let us say, at one time to the employment in an aircraft factory of 50 per cent of wood workers and 30 per cent of steel workers, and there is no human authority who can with sanity pretend to say that in some brief time these proportions will, or will not, be widely changed. If steel or aluminum were to form a greater part of the structure than they do now, because of safety, or diminished fire risk, or what not, any agreement with the wood workers' union for continuity of their employment would install them as a paid but non-working body of pensioners on the industry. Alternatively, British aircraft, hampered by such an agreement, would remain locked in its old technical groove and be debarred from adopting the devices and inventions which throughout the rest of the world would be improving the machine. Surely a hopeless prospect for a technical service required for war!

It is clear, then, that the agreements between manual and brain workers on aircraft must be part of the general agreement on which our new millennium is to be based.

TO SAVE DISAPPOINTMENTS.

Employment of Soldiers and Sailors.—The industry will, even when the most strenuous efforts have been made to save it by such continuance of air service orders as are at all possible, and when such aerial transport work is started as may be decided on, still be cut down drastically. Hence very great disappointment will be spared to many a good man by realizing thus early that there will be no expansion of employment on aircraft which will absorb them wholesale, but rather a manifold shrinkage.

PUBLIC CONTACT.

Accidents.—It is unfortunate that the chief contact which the public of England makes with aeronautics is in relation to accidents and casualties.

The achievements appear, it is true, but it is almost impossible to visualize them save perhaps in the case of an occasional flight of unusual distance, such as the recent 3,000-mile flight from London to Constantinople, or between London and Turin, and these things appear to be disconnected items giving no impression of the hundreds and thousands of miles of continual flying which is taking place. These show flights are few, not because they cannot be frequently achieved, but because war does not provide for show and *réclame*. The public cannot by mere lecture be brought to the point of view which is standing with those who have been fully familiar with aircraft; we are so imbued with its utility and versatility that we are no more discountenanced by occasional breakages than we are deflected from hiring a taxi by seeing a sideslip against the curb. Those actually engaged in design, who naturally need to use every element of experience which accidents may bring to enable them to perfect their work, are anxious to study and analyze these occurrences.

REASONS FOR ACCIDENTS.

Air accidents can be divided into groups for the purpose of their study, and when we proceed so to divide them we find that by far the largest category includes those which are due to the two following circumstances: (1) The absence of landing-grounds distributed in easy stages to which a flier can turn for refuge if his engine should stop; (2) the inexperience of fliers themselves, since the majority of the experienced have been drawn off to the war. If we remove from the records of accidents, other than tuitional, all those which could have been avoided were these causes non-existent, the residue would be less significant in *fact*, and vastly less significant than is the *impression* abroad. Beyond this it is still reasonable and advantageous to subdivide the causes of this residue of accidents into classes, so that attention should be drawn to that which may be eliminated, be it in design or in handling.

Of the remainder, the most important is what is commonly called "Pilot's error." This does not imply that he is blameworthy. In war, maneuvers which are definitely hazardous are necessary; people may be surprised to hear that to loop-the-loop is a useful war maneuver which may greatly puzzle an antagonist in the air, since the looped path takes the place of the expected forward movement, and therefore falsifies the aim of the adversary, or throws out all his estimate of the direction in which to fly for purpose of intercepting him. Another maneuver, known as "spiraling," or "spinning," if contrived to give the impression that the airplane has in fact been thrown out of control, is one of many ruses for breaking off a fight. The learning of these maneuvers must be achieved before they can be used in war, and tends to swell the number of broken machines. Rapid diving, such as might be made in an attack on observation balloons, may similarly be a cause of accident until judgment is acquired in the method of "flattening" out the dive. Even to describe a *very* small circle in a horizontal plane implies unusual stress on an airplane, if accomplished at high speed. None of these maneuvers are called for in trader work, while, to add to the comparative safety of the peace aircraft, we can appreciate the effect of removing the war demand for high performance and speed. Under fire and pursuit safety comes from lightness and maneuverability, as much as, or more than, it does from strength and solidity; in trader work the conditions are largely changed, if not reversed.

REDUCED RISKS.

Aircraft travel and transport therefore will, if landing-grounds are provided, not be exposed to any of the risks above named, and accordingly from the list of accidents which occur today we must eliminate almost all

these in any prognostication as to the future safety of flying. Before such an audience I need not belabor the so-called *air pocket*. It never did exist, and generally there seems to be little probability of any airplane, even when designed on the factor of strength used today, with the limited science of today, being broken in the air under the conditions of aerial travel.

Much need not be made of the fire dangers on aircraft, for when the matter is regarded dispassionately, the fire risk by actual experience is small. Of the fires which have occurred since thousands of airplanes have been turned out per month, the large majority have arisen when the airplane has struck the ground and the petrol vapor from the broken-up tanks has come into contact with a spark. Now, this breaking-up of a machine on the occasion of a forced landing again becomes a rare occurrence if the line of landing grounds suggested for aerial routes is provided. Most of these smashes, some fraction of which have resulted in fires, have occurred by reason of the flier holding up his machine unduly long in his search for a suitable place to alight in a country where no provision for alighting exists.

INCREASING SAFETY.

Of the mechanical precautions taken to avoid fire, we know that already it has been possible to make a magneto which, when filled with explosive vapor and surrounded by it, gives rise to no ignition of the mixture. We have also learnt that the back-firing or "popping" of an engine into the induction pipe, which throws a flame back, can be made innocuous by drawing the air supply of the engine from outside the body work. This, indeed, is standard practice today. The use of electrically heated clothes, called for by fighting at heights such as 20,000 feet, will not be nominally desirable, and as for appliances for wireless telegraphy, which will have their place for giving trader craft their direction and for calling to the ground, they will not under peace conditions impose any risk of fire which cannot be circumvented. In general, the standard of safety from this point of view will be automatically enhanced in all transport work. Carelessness, such as makes people bring matches into a T. N. T. factory, cannot be expunged from the human race, but this class of occurrences need not disconcert us at all. In filling up with petrol there is the chance of spillages, and no doubt but that every aerodrome should be equipped with portable fire-extinguishers of light weight made available at filling points, and no doubt also but that all large airplanes will carry one in an accessible position, until the proved absence of utility causes them to be relinquished.

FOG DANGERS.

The accident, if it can be so called, of losing one's way in a fog will be far less likely to result in disaster when, to quote it once more, there exists a multiplicity of landing grounds, because, on the one hand, no fog has been found to extend more than a very limited height, say, 700 yards maximum, and, on the other hand, because we now know that fogs are quite local in their occurrence at any one moment. A befogged flier, instead of alighting through a foggy patch, merely moves a little farther on before landing. This does not imply that projects for signifying to a flier who is above the fog the correct position of his aerodrome by pilot balloon or raised lights are to be discarded, but this is not the place to deal with that point.

PARACHUTES.

Parachutes are spoken of, but by many fliers are not thought particularly desirable. We know that at present they require a height of some 500 feet to open out and afford the safety which they appear to offer, and it is

usually below these levels that the flier becomes convinced he is to be exposed to some risks, say, by the conditions of the ground together with the stopping of his engine which causes him to alight. One would have to be very seriously out of touch with those who fly daily and really know their job if one continued in the impression that numbers of accidents are the inevitable concomitant of aerial travel, and it is sufficient to say that any such opinion may be dismissed as one which has arisen from the peculiar conditions of Press publicity in war and the exclusively high-pressure and high-performance conditions of the development of aeronautics up to today. The lack of contact of the public with the serious and successful work which has been achieved has already been mentioned.

WIND TUNNELS.

Research.—It is possible that someone will question the need for research and for State endowment thereof; it may be said that we are well on and can now trust to practical developments, since root-and-branch retrenchment of expense is called for. This is a complete *non sequitur*. If cheap progress is wanted, the cost of progress is enormously reduced by making and testing models two feet wide instead of making structures of a 100-foot span. The ratio of cost is as £10,000 to £50, and it is the £50 scheme which I am advocating. I cannot give a list of researches, but ask you to consider the following examples. We need the scale corrections from model parts to full size; we need to test propulsive mechanisms, without building them; we must know the value of new wing sections; of reflexed wings; of hinging the flaps for controls; we must, in order to get any airplane stable, know the movement of the center of pressure on the wings when the attitude of flight is varied; we must study the effect of vibration on structural parts, etc.

We are aware that the great utility of aerial travel is speed, that the great enemy of speed is head resistance, and we must be able by simple model tests to forecast the speed which will be attained by a given design. We must be able to know, before we introduce a variation in design, the amount of economy of resistance which can be made by altering structural parts. All this class of work can be effected in the wind tunnel, and already most serious aeronautical construction firms are equipping themselves with a wind tunnel. Probably the greatest value of such equipment is its educational effect. The wind tunnel has fought its way to the front, in spite of much opposition, and it is still to be recognized that unless its indications are interpreted by skilled persons grave blunders will be made, but this in no wise detracts from its utility when properly employed.

ENGINES, AIR SCREENS AND OTHER THINGS.

We are aware that the thermal efficiency of engines employed on aircraft is as high, in spite of their small size and weight, as the efficiency of Diesel engines giving thousands of horsepower. Yet this figure is only 30 per cent, and by research alone can we hope to obtain an extension of this percentage. The study of airscrew efficiency, the effect upon it of the body, the means for making airscrews smaller than they are at present, and thereby simplifying the whole outline of an airplane, is a matter which research must tackle. The use of new alloys, of new methods of totally enclosing the flier in an airplane, new methods of housing the landing gear, so as to get rid of its head resistance, the protection of the fabric of the wings from disintegration by ozonization due to actinic light, all these and countless more matters can only be economically and expeditiously studied in laboratories equipped for the purpose, they will effect enormous economies in aeronautical upkeep, and aero research must be pressed on and financed after the war.

Broadly, it may be said that air transport will not develop save by tak-

ing a line of footprints given by research. To say that we want it does not by any means ensure that it will proceed apace as it should. Research of various kinds is remunerative in two different ways: (a) quickly and (b) eventually. Those studies which conduce to (b) rarely produce (a), a quick return. On the other hand, researches which afford us (a) are very likely, indeed, to be remunerative both quickly and eventually.

If we now subdivide the various researches which might forward aeronautics into two other classes—those (a) which result in some proprietary advantage to the inventor, and those (b) which have a general effect on progress, and which are not susceptible of being privately protected—we find that there is every inducement to the research worker to concentrate upon the class of improvement which I have called (a) in each of the above pairs of groups, that is to say, the device which can both be made proprietary and which is remunerative quickly.

This must, unless we are very foreseeing and courageous, operate to deflect research workers from the other and frequently much more important class of study, viz: those which are of *general* application and those which are eventually of great money value to the whole of the industry, whether of construction or of transport.

Patent laws stimulate advancement by providing that there shall be a reward, and that all and sundry shall not be able to possess themselves of the fruits of one man's labor, and thereby dispossess him. Means must be devised for securing that the British industry, as a whole, shall not be tempted to exclude from its study the great groups which I have classed as (b), which are so frequently not protectible by patents; and this business falls naturally to the new Department of Scientific and Industrial Research, who will, no doubt, find the hearty cooperation of the Society of British Aircraft Constructors and the Aeronautical Society. It is for the public to see that the determination toward progress exists, and then the outcome will first of all be the necessary funds, and thereafter the harvest of results and economies.

The Air Board has formed a Civil Aerial Transport Committee, which is engaged in considering these and most of the other matters dealt with in any such discussion as this, and if they recommend active steps to be taken, let us hope that there will be firm public support for any expenditure and assistance which shows proper foresight, always with the considered approval of the new Air Minister.

AIR ROADS.

Routes.—One of the things wanted now is some inquiry made experimentally by actual flight for the purpose of determining the best aircraft trader and postal routes which may be useful and eventually payable. London-Paris has been suggested, with extension to Marseilles, Turin and Egypt. Links between capitals have also been spoken of, and links with India and South Africa, etc. Short runs will probably come first. In all short-distance runs the problem is complicated when the existing rail and steam facilities compete. There is a marked advantage for aircraft every time that transshipment of any kind occurs on the ordinary means of transit. This favors London-Paris and London-Dublin, for example. Similarly, and for similar reasons, whereas no great advantage might be gained by a London-Glasgow route, where the train travels at fifty miles per hour, and the journey can easily be done during the lost hours of the night without the traveler being roused from his sleeping berth, we shall find that cross-country journeys, those involving changes of train, waiting at stations for connections, will be instantly eclipsed in comfort and speed by aerial travel and aerial mail deliveries. One could suggest Cardiff and Newcastle, or the like.

I do not think that much is to be gained by suggesting routes at the present moment, unless we propose to permit a few machines to be with-

drawn from war service, and then, by using a few pilots who are for one reason or another not fit for war service, explore the practical possibilities of such transport. I think it far better to start by carrying newspapers and letters and packets, rather than passengers at first, for every reason. The routes will be unknown to the pilots, their experience may perhaps not be great—the advertisement cannot usefully be made in war time, and, generally, it is not desirable, save in exceptional cases, to expend petrol for personal travel, or to expend money in adapting designs to passengers' comfort. Yet if this comfort were omitted, a false impression of the facilities to be afforded would easily get spread abroad.

I do not know whether certain types, whose performance is below the ever-rising requisites for war, could be made available, but I should have thought that with good will they could be managed, and, if so, the experiments made in war-time would help us to start *quickly*, on declaration of peace, that which might eventually help to create orders—only a few, it is true, but still a small fraction of what is wanted for the maintenance of the production factories.

FUTURE SPEEDS.

Speed.—I am inclined to endorse Mr. Holt Thomas's view that, if we consider the average wind speed as 30 miles per hour, sometimes with us and sometimes against, we shall need machines whose own air speed is 120 miles per hour, in the case when there is any competition with other means of transport.

On cross-country journeys far lower speeds will be vastly faster than the existing systems, and will at once command attention.

As we know, 120 miles per hour is a very ordinary speed today for aircraft, and offers no technical difficulties in alighting—indeed, Captain Green's interesting forecast, under certain conditions which he laid down, of an airplane to travel at 240 miles per hour, did not appear to raise any feeling of doubt or hesitancy at the Aeronautical Society about three weeks ago—though a similar audience in 1911 was very skeptical about the advisability of introducing, even for military purposes, my proposed speed of 72 miles per hour. This gives us an idea of the changes which have gone on in men's minds by legitimate extrapolations from the actual achievements with which they are familiar.

I have nearly finished, but before doing so I must once more reiterate that the most hopeful estimate for immediate aircraft transport and travel does not save the situation. There must be governmental action to maintain the country's productivity and designing capacity for military and naval aircraft in the first place, and as the mainstay of the country's factory organization.

A NEW ARENA.

In conclusion, it is only half a truth to say, with Frederick List, "The sea is the high street of the earth. The sea is the parade ground of the nations. The sea is the arena for the display of strength and enterprise of all nations." There is now the air. A great instrument of power tending to peace and usefulness and good will can be designed in a week, if only a British, a French, an American, an Italian plenipotentiary could be empowered to sit around a table. That instrument is the aerial way. Its regulations must not be founded now upon speculations; they must be evolved later upon experience.

All the Allied nations desire the same thing, all are agreed that just as in time of peace we must prepare for the catastrophe of war—so in war we must prepare our thoughts and plans for the cataclysm of peace. The word is used advisedly, for a very sinister situation will develop in the world of aircraft during the hiatus which first was publicly indicated, I think, by myself, in the discussion on Mr. Holt Thomas's paper before the Aeronautical Society, and which Lord Cowdray himself promised to make every effort to bridge.—"Aviation."

BOOK REVIEWS.

INTERNAL COMBUSTION ENGINE MANUAL, by LIEUTENANT COMMANDER F. W. STERLING, U. S. N. Published by R. BERESFORD, Washington, D. C., 168 pages, 93 illustrations. Price, $2.00.

This Manual has again been revised and brought up to date, to keep step with the recent advances in internal combustion engines, and although originally written for use as a text book for a short course on internal combustion engines for the Midshipmen at the Naval Academy, has now arrived at a shape where it is quite suitable for courses such as are being offered by many of the universities of the country for the training of men for handling of motors in aerial work and in marine service, as it covers the theory and practice of the internal combustion engine and allied subjects, without the elaborate mathematical demonstrations and theories found in most text books.

The manual is arranged in a logical order, the subject of fuel being taken up first, this being the element that governs the design and operation. Following this the fuel systems, ignition system and lubrication systems (which have been enlarged) and cooling systems are taken up in detail. Various methods of governing and information as to operation to be gained from indicator cards is covered, also operation, troubles and remedies.

Various types of engines are then discussed, such as the standard engine for Navy submarine chasers, Mietz and Weiss Marine engine, the Knight sleeve valve motor, etc.

The chapter on Diesel engines has been largely rewritten and a new chapter on aerial motors added, illustrated with half tones of the latest types, such as Hall-Scott, Sturtevant, Ashmusen, Curtiss and Gnome.

RAPID TRAINING OF RECRUITS—A PRACTICAL METHOD, by 1ST LIEUT. M. V. CAMPBELL, FIELD ARTILLERY, N. A. 180 pages, limp cloth binding. Price, $1.00. Published by FREDERICK A. STOKES Co., New York.

This little volume is a practical scheme for quick results in training the National Army, based on the definite record of what has been actually accomplished by England.

The aim is to give the recruit instructor in American cantonments a thoroughly tested plan for whipping his men into shape speedily. A typical day's work is minutely outlined, lectures are suggested, in fact every detail necessary to an intensive program is fully treated.

The book is fully as useful to the recruit himself as to his instructor. Any intelligent enlisted man, eager for advancement, will find it of incalculable value in his efforts to get ahead.

THE MARVEL BOOK OF AMERICAN SHIPS, by CAPTAIN ORTON P. JACKSON and MAJOR FRANK E. EVANS. Publishers, FREDERICK A. STOKES COMPANY, New York. Price, $2.50.

A reliable and fascinating book for all who are interested in the sea. Major Frank E. Evans, United States Marine Corps, and Captain Orton P. Jackson, U. S. N., authorities on naval matters, describe in detail every kind of ship—the Man-O'War's-Man, Superdreadnought, Submarine, Destroyer, Light-Ship, Liner, Yacht and Merchantman.

The reader is taken to the great ship yards, where our huge sea-going steamers and fighters are constructed, and introduced into the secrets of their building. There are vivid accounts of great sea battles, gun firing and signalling, deep sea diving, and countless other subjects connected with the sea.

For those not versed in the lore of the sea it is most interesting and instructive. For those that go down to the sea in ships the pictures take one from "The Days of Wooden Ships and Iron Men" to the "Landing at Vera Cruz."

Probably the most notable feature of this book is its abundance of illustrations—there are twelve plates in full color, and over 400 illustrations from photographs. Especially interesting is the chart of the flags used in the signal code, and that showing the types of funnels and company flags by which the ships of the various great steamship lines can be recognized. Altogether a book that one will want to read from cover to cover.

MILITARY AND NAVAL RECOGNITION BOOK. LIEUT. J. W. BUNKLEY, U. S. N. 51 full page plates, 18 in colors, $1.00. D. VAN NOSTRAND COMPANY, New York. 224 pages. Cloth, Khaki.

An extremely useful manual, compact in form, lucid in style, easy to understand, with a good index and sufficiently well illustrated. The Organization of various branches of the Service, together with uniforms and insignia, are well treated.

Written by an officer who is familiar with his subject and one who has apparently made diligent use of works of reference. The work shows painstaking research and patient collection of material which is authoritative.

Useful for men in service "over here" or "over there." More useful for Reserve Officers and newly enlisted men. Non-technical in style, so that civilians may readily comprehend its scope, and also learn with facility the various insignia, customs and usages of all branches of the Service.

Form and press work reflect credit on its publishers.

FINDING AND STOPPING WASTE IN MODERN BOILER ROOMS. Published by the HARRISON SAFETY BOILER WORKS, Philadelphia, Pa. Price $1.00. 274 pages, limp cloth binding.

This handy volume is a reference manual to aid the owner, manager and boiler-room operator in securing and maintaining plant efficiency. It is replete with charts, tables and references of unusual value to the combustion engineer. Its five sections

treat of (1) fuels, (2) combustion, (3) heat absorption, (4) boiler efficiency—boiler testing, (5) boiler plant proportioning and management. A very meaty book for anyone interested in the subject.

ARMY AND NAVY UNIFORMS AND INSIGNIA. HOW TO KNOW RANK, CORPS AND SERVICE IN THE MILITARY AND NAVAL FORCES OF THE UNITED STATES AND FOREIGN COUNTRIES. By COLONEL DION WILLIAMS, U. S. M. C. 8 Illustrations in color and 117 in black and white. FRED'K A. STOKES Co., New York. 302 pages $1.50. Bound in blue cloth.

This is a work in manual form but full of information in extremely compact shape. Uniform and insignia are treated historically, as well as flags, rank and corps. Col. Williams evidently knows his subject. His style is clear, easy to understand because it is not technical, and attractive. The illustrations, which include photographic reproductions, are particularly interesting.

It is a valuable work, useful to those in the Service as well as to civilians. Reserve Officers should find it very essential in securing quickly a knowledge of necessary information. The Chapters on Foreign Uniforms and Honors and Distinctions are of great value. Paper, letter press and type are good.

ASSOCIATION NOTES.

A meeting of the Society was held on Saturday, December 29, 1917, for the purpose of counting ballots cast for officers of the Society for the year 1918, and for and against amendments offered to the By-Laws.

The following were declared elected officers of the Society for 1918:

President, Captain H. P. Norton, U. S. Navy.

Secretary-Treasurer, Lieut. Commander F. W. Sterling, U. S. Navy (retired).

Council.

Captain T. W. Kinkaid, U. S. Navy.

Naval Constructor Robert Stocker, U. S. Navy.

Lieutenant Commander S. M. Robinson, U. S. Navy.

The amendment of By-Laws was decided upon by a vote of 252 to 10.

FINANCIAL STATEMENT, 1917.

Publication Statement.

The following statements show the returns from and the cost of the publication for the JOURNAL of the Society for the year 1917:

RETURNS.

Dues	$5,601.21
Subscriptions	2,089.25
Sales	1,110.10
Advertisements	2,530.48
Exchange	3.40
	$11,334.44

COSTS.

Printing	$4,518.47	
Engraving	529.85	
Drafting	8.50	
Manuscript	1,183.25	
Postage	140.70	
Commission and discount	202.66	$6,583.43
Gross profit		$4,751.01

GENERAL PROFIT AND LOSS STATEMENT.

The following statement shows the result of the business of the Society for the year 1917 as a whole:

	Loss.	*Profit.*
Gross profit from publication		$4,751.01
Interest		398.58
		$5,149.59
General expense	$1,611.35	
Banquet	65.08	
Current profit and loss*	1,483.79	3,160.22
Net gain		$1,989.37

BALANCE STATEMENT.

The following statement shows the difference between the assets on January 1, 1917, and those on December 31, 1917.

Assets December 31, 1917.		
Cash	$4,282.24	
Accounts receivable, advertisements	1,881.42	
Accounts receivable, dues	1,001.25	
Accounts receivable, sales	61.00	
Accounts receivable, subscriptions	246.10	
Army and Navy Club bond	1,000.00	
Washington Railway and Electric bonds (6)*	4,320.00	
Liberty Loan bond	1,000.00	
Certificate of Deposit, Munsey Trust Co.	500.00	
Furniture	51.84	
Miscellaneous asset (gold content of medal)	16.00	$14,359.85
Total assets January 1, 1917		12.370.48
Net gain		$1,989.37

*Current profit and loss consists mainly of readjustment to the market value of W. R. and E. bonds, and of over $500.00 of old accounts due for a period of three to five years, which it was thought advisable to write off at this time. Since purchase the W. R. and E. bonds have been carried at cost instead of market value, and it is believed a more satisfactory method of accounting is to carry them at the market value, which amount could be realized if a sale became necessary.

Respectfully submitted.

F. W. STERLING,
Secretary-Treasurer.

Audited and found correct.

S. M. ROBINSON,
H. T. DYER,
F. W. STERLING,
Audit Committee.

The numerical strength of the Society on December 31, 1917, was:

Active members	707
Associates	407
Subscribers	422
Exchanges	77
Total	1,613

The following Members and Associates have joined the Society since the publication of the last JOURNAL:

MEMBERS.

Church, V. S., Ensign (T.), U. S. N.
Coddington, P. L., Ensign (T.), U. S. N.
Conally, James M., Ensign (T.), U. S. N.
Crocker, W. S., Ensign (T.), U. S. N.
Cruze, J. Roger, Lieutenant, U. S. N.
Davidson, L. H., Ensign (T.), U. S. N.
Dunn, M. L., Ensign (T.), U. S. N.
Eaton, L. F., Ensign (T.), U. S. N.
French, H. W., Ensign (T.), U. S. N.
Herd, Clifton B., Ensign (T.), U. S. N.
Murray, A. D., Ensign (T.), U. S. N.
Pick, Robert B., Assistant Naval Constructor (T.), U. S. N.
Sutcliffe, Marcus A., Ensign (T.), U. S. N.
Walker, Richard L., Ensign (T.), U. S. N.

ASSOCIATES.

Denslow, A. C., Room 512, 61 Broadway, New York City.
Engel, H. W., Lieutenant, N. N. V.
Eye, Theodore, Jr., Machinery Drafting Room, Navy Yard, Charleston, S. C.
Ferrier, George B., Land Title Building, Philadelphia, Pa.
Fletcher, W. D., 36 Cushing St., Dover, N. H.
Gaehr, David, 952 Rockefeller Building, Cleveland, Ohio.

Gross, Charles F., Instructor, Naval Academy, Annapolis, Md.

Hughes, Charles H., Naval Architect and Engineer, 27 William St., New York City.

Kiley, D. L., Navy Yard, Norfolk.

King, James B., Machinery Division, Drafting Room, Navy Yard, Charleston, S. C.

Lisle, Thos. Orchard, South Ferry Building, New York City.

Lynes, Gorham M., Lieutenant, U. S. N. R. F.

Parish, William F., Ontario Apartments, Ontario Road, Washington, D. C.

Rust, Elmer P., Lieutenant, U. S. N. R. F.

Thomas, David D., Baltimore S. B. & D. D. Co., Baltimore, Md.

Willoh, R., Lieutenant, U. S. N. R. F.

Wilson, David H., Jr., 52 Vanderbilt Avenue, New York City.

ADDRESSES.

The Secretary renews his recent request that all members keep him advised as to their addresses in order that JOURNALS may be delivered promptly.

AN ASSOCIATE of the Society offers for sale a complete set of the Journal, Vols. I to XXIX, all in good condition. Vols. I to XXV are bound. Price for bound volumes $6.50 each; for unbound volumes $5.00 each. Address A. H. R., this office.